Pump fundamentals

2

Nomenclature

a	piston rod cross-sectional area, mm^2 (in^2)
A	plunger or piston cross-sectional area, mm^2 (in^2)
B	bulk modulus of fluid, kPa (psi)
BHP	brake horsepower, kW (hp)
C	constant (for type of pump)
d	displacement per pumping chamber, m^3 (gal)
D	pump displacement, m^3/h (gpm)
d'	pump piston or plunger diameter, mm (in.)
d_w	diameter of wrist pin, mm (in.)
E_M	pump mechanical efficiency
f_p	pump pulsation frequency, cycles/s
g	acceleration due to gravity, 9.81 m/s^2 (32.2 ft/s^2)
H_A	the head on the surface of the liquid supply level, m (ft)
H_{AC}	acceleration head, m (ft)
H_f	pipe friction loss, m (ft)
HHP	hydraulic horsepower, kW (hp)
H_p	total head required for pump, m (ft)
H_{PH}	potential head, m (ft)
HSH	vapor pressure head, m (ft)
H_{VH}	velocity head, m (ft)
H_{VPA}	static pressure head, m (ft)
K	a factor based on fluid compressibility
L	length of suction line, m (ft)
l	length of wrist pin under load, mm (in.)
m	number of pistons, plungers, or diaphragms
n	stroke rate or crank revolutions per min, rps (rpm)
NPSH	net positive suction head, m (ft)
$NPSH_A$	net positive suction head available, m (ft)
$NPSH_R$	net positive suction head required, m (ft)
P	bladder precharge pressure, kPa (psi)
P_c	plunger load, N (lb)
PL	pressure increase, kPa (psi)
Q	flow rate, m^3/h (ft^3/s)
q	flow rate, m^3/h (gpm)
Q'	flow rate, m^3/h (BPD)
S	stress, kPa (psi)
s	stroke length, mm (in.)
S'	valve slip, percent
(SG)	specific gravity of liquid relative to water

Surface Production Operations. https://doi.org/10.1016/B978-0-12-809895-0.00002-8

V	velocity in suction line, m/s (ft/s)
Vol	volume of surge tank, m^3 (ft^3)
$(\mathbf{Vol})_g$	required gas volume, m^3 (ft^3)
Z	elevation above or below pump centerline datum, m (ft)
ΔP	static pressure, kPa (psi)
ρ	density of fluid, kg/m^3 (lb/ft^3)

2.1 Engineering principles

2.1.1 Background

Pumps serve many purposes in production facilities. The largest pumps are usually shipping or pipeline pumps which increase the pressure of oil or condensate so that it can flow into a pipeline or be loaded into tankers, barges, railroad cars, or trucks. Large pumps are also used with water injection systems for disposing produced water or for waterflooding. Smaller pumps are used to pump liquids from low to higher pressure vessels, to pump liquids from tanks at a low elevation to tanks at a higher elevation, or to transfer liquids for further processing. A production facility's utility system often has many pumps, which may be used for firewater wash down and utility water, heat medium, fuel oil or diesel, and hydraulic systems.

The engineer must be able to select the proper pump for each application, determine horsepower requirements, design the piping system associated with the pump, and specify materials and details of construction for bearings, seals, and so on. On standard applications the engineer may allow the vendor to specify materials and construction details for the specified service conditions. Even then, the engineer should be familiar with different alternatives so that he or she can better evaluate proposals and alternative proposals of vendors.

There are a number of factors that should be considered when selecting a pump for a specific application. The primary goal is the same, that is, providing a pump that maximizes company profits (low cost) while providing safe, reliable (trouble free) equipment that satisfies operating requirements and meets local environmental constraints. Selecting the right pump is difficult due to the number of different kinds of pumps available.

Meeting the primary goal involves optimizing the following primary factors:

- Minimizing the initial pump cost
- Meeting safety and environmental concerns
- Reducing installation and commissioning expenses
- Reducing maintenance expenses
- Maximizing reliability
- Starting up on time
- Maintaining maximum production

Sound engineering judgment is important when deciding which of the listed factors are the most important.

2.1.2 Types of fluids

Pumps are used to move fluids. Fluids include liquids, dissolved gases, and solids. Gases include dissolved air and hydrocarbon vapors. Solids include sand, clay, and scale. Most common types of fluids pumped in upstream operations are water, crude oil, condensate, lube oils, glycols, and amines.

Each fluid has different physical properties. Physical properties that must be taken into consideration when sizing and selecting a pump are as follows:

- Suction temperature
- Specific gravity
- Viscosity
- Vapor pressure
- Solids content (if predictable)
- Lubricity (slipperiness)

2.1.3 Pump classification

A pump can be defined as "as mechanical device that adds energy to a liquid for the purpose of increasing its flow rate and static pressure." Pumps are divided into two major categories: positive displacement and kinetic energy. These two categories are further divided into numerous subdivisions.

2.1.3.1 Positive displacement pumps

Positive displacement pumps add energy to a fluid by applying force to the liquid with a mechanical device such as a piston or plunger. A positive displacement pump decreases the volume containing the liquid until the resulting liquid pressure equals the pressure in the discharge system. That is, the liquid is compressed mechanically, causing a direct rise in potential energy. Most positive displacement pumps are reciprocating pumps in which linear motion of a piston or plunger in a cylinder causes the displacement. In rotary pumps, another common positive displacement pump, a circular motion causes the displacement. There are several manufacturers of positive displacement pumps which are often found in high-pressure services. As shown in Fig. 2.1, positive displacement pumps are classified as either:

- *Reciprocating*—use pistons, plungers or diaphragms to displace fluid.
- *Rotary*—operate through the mating action of gears, lobes, or screw-type shafts.

2.1.3.2 Kinetic energy (dynamic) pumps

Kinetic energy (dynamic) (energy associated with motion) is added to a liquid to increase its velocity and, indirectly, its pressure. Kinetic energy pumps operate by drawing liquid into the center of eye of a rapidly rotating impeller. Radial vanes on the impeller then throw the liquid outward toward the impeller rim. As liquid is thrown outward, more liquid is drawn into the resulting low-pressure area in the eye through a suction port in the pump casing. As liquid leaves the impeller, it comes into contact with the pump casing or volute. The casing is shaped to direct the liquid

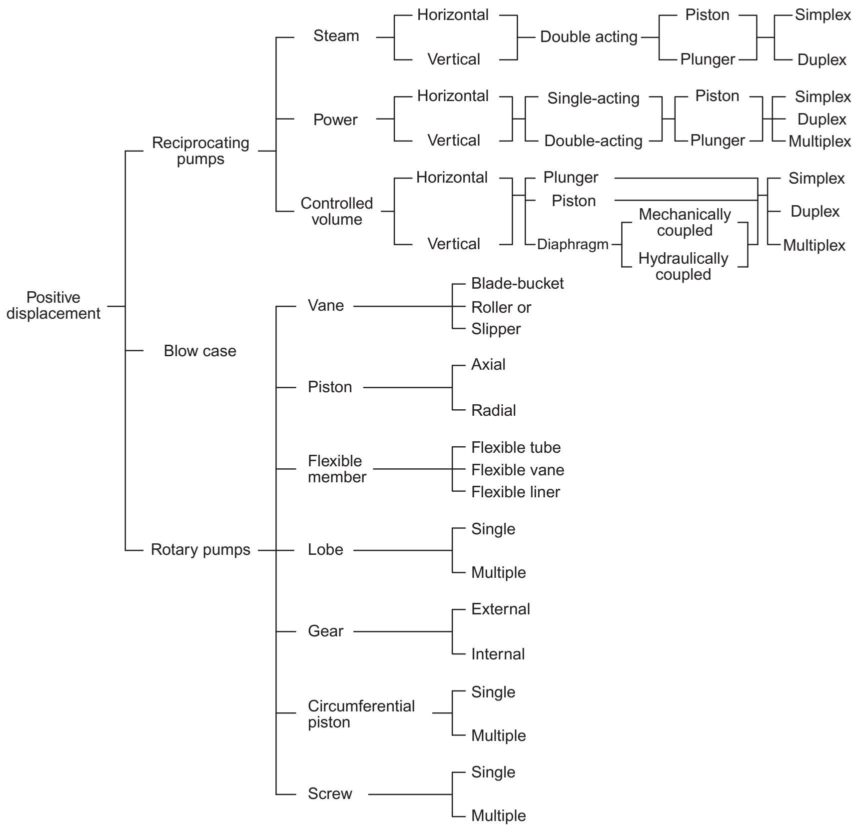


Fig. 2.1 Positive displacement pump classifications.
Courtesy of Hydraulic Institute.

toward a discharge port. The casing slows the liquid and converts some of its velocity into pressure. As shown in Fig. 2.2, kinetic energy pumps are classified as either:

- *Centrifugal*: Includes radial, mixed, and axial flow designs. They account for over 80% of pumps used in production operations because they exhibit uniform flow, free of low frequency pulsations and are not subject to mechanical problems.
- *Regenerative turbine*: Often times the regenerative turbine pumps are considered to be positive displacement type pumps. They are available in single stage and multiple stages, for low flow/high-pressure applications.
- *Special effects pumps*: Reversible centrifugal pumps and rotating casing (pilot) pumps fall in this category.

Fig. 2.3 illustrates pumps that are commonly encountered in production operations. Pumps are commonly rated by

- Flow rate
- Differential pressure/head

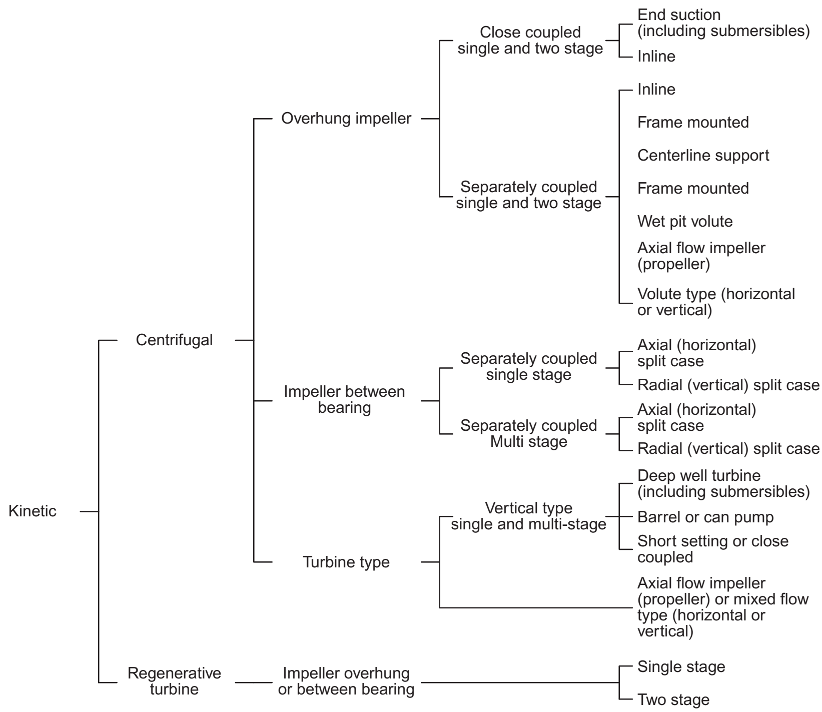


Fig. 2.2 Kinetic energy pump classifications.

- Horsepower
- Suction pressure

2.1.4 Positive displacement pumps

Positive displacement pumps, although not as common as centrifugal pumps, are also widely used in production operations. The three major positive displacement subdivisions are reciprocating, rotary, and controlled volume (metering). The reciprocating pumps use pistons, plungers, and diaphragms to displace the fluid while rotary pumps use vanes, gears, screws, lobes, and so on.

2.1.5 Centrifugal pumps

Centrifugal pumps are by far the most widely used pumps and have been estimated to make up 80% of all pumps used in production operations. They are widely accepted because they combine a relatively low cost with high reliability, compact size, non-pulsating flow, and easy maintenance. They are also widely available, cover a broad flow/pressure application ranges, and can operate over a wide flow range.

Centrifugal pumps are usually purchased to meet one of two levels of duty.

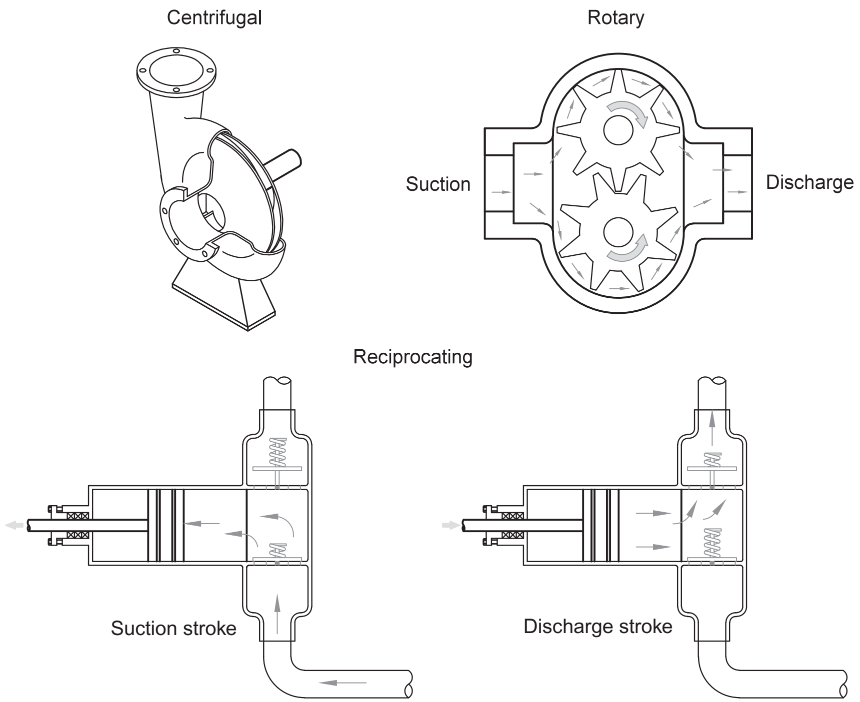


Fig. 2.3 Pumps commonly encountered in production operations.

Medium Duty pumps are used in general, noncritical, nonhazardous service. These pumps are usually built to American National Standards Institute (ANSI) Standard B73.1 “Horizontal End-Suction” or ANSI Standard B73.2 “Vertical In-Line” specifications.

Heavy-duty pumps are used in critical, hazardous, or “heavy-duty” service including chemicals, refining, and other producing services. These pumps are usually built to American Petroleum Institute (API) Standard API 610 “Centrifugal Pumps for General Refinery Services” specification. Centrifugal pumps used as Fire Pumps are most often heavy-duty pumps.

2.1.6 Miscellaneous pumps

Several other types of pumps have been designed to accommodate specific needs. These pumps include

- Air diaphragm pumps
- Regenerative turbine pumps
- Jet pumps
- Slurry pumps

Sucker rod and electric submersible pumps (ESP) pumps are both very common and important in production operations. These pumps are discussed in Chapter 6. Progressive cavity pumps are covered in the Rotary Pump discussion in Chapter 5.

2.1.7 Pumping system design

The process of designing any pumping service involves three major activities:

- Process design
- Mechanical design
- Vendor specifications

2.1.7.1 Process design

- *Obtain flow rate*. Define any flow variation that should be included in design, such as start-up conditions, future expansion, and maximum flow. Select a value for rated flow rate. Convert the required rated flow rate at pumping conditions into US gallons per minute.
- *Determine the liquid properties critical to pump design*. Properties of importance include specific gravity, temperature, viscosity, pour point, and so on. Values are required at pumping conditions and, in some cases, at ambient conditions.
- *Calculate available suction conditions such as rated suction pressure, maximum suction pressure, and* $NPSH_A$.
- *Determine the effect of the selected control system on pump performance requirements (i.e., constant bypass, backpressure control valve, or variable speed).*
- *Calculate the minimum discharge pressure requirement of the pump*.
- *Calculate the total dynamic head (TDH) at the specific gravity corresponding to rated pumping temperature*.

2.1.7.2 Mechanical design

- Determine the design pressure and temperature required for the pump and its associated piping.
- Select pump type and driver type.
- Select materials of construction.
- Determine sparing (backup) requirements and the need for parallel operation.
- Determine other installation requirements, such as control system details, auto-start of standby pump, and so on.
- Select shaft seal type and determine the requirements for an external flushing or sealing system.
- Estimate utility requirements.
- Document the design: calculations, studies, design specification text, utility requirement estimate summary, and so on.

The following factors listed have a significant influence on the mechanical design and investment cost.

- Number of pumps installed (in parallel)
- Type of casing material
- Net positive suction head available ($NPSH_A$)
- Total dynamic head (TDH) requirement

- Flow rate per pump in gallons per minute, gpm (m^3/h)
- Design pressure and temperature
- Type of pump selected
- Corrosiveness/toxicity of the fluids
- Solids content of the fluids
- Power requirements
- Type of driver selected

2.1.7.3 Vendor specifications

Factors that have the greatest influence on the selection of the most cost-effective pump type include

- Capacity, gpm (m^3/h)
- Total dynamic head (TDH)
- Maintenance
- Viscosity
- Capacity control

Within the general type selections, a particular construction style is most influenced by

- Discharge pressure
- $NPSH_A$
- Fluid temperature
- Space and weight limitations

2.2 Hydraulic principles

Hydraulics deals with the mechanical properties of water and other liquids and the application of these properties as they apply to engineering. Hydraulics is divided into the following two areas:

- Hydrostatics (fluids at rest)
- Hydrodynamics (fluids in motion)

2.2.1 Hydrostatics

2.2.1.1 General considerations

A liquid has a definite volume when compared to a gas, which tends to expand indefinitely. When unconfined, a liquid seeks the lowest possible level. Due to its fluidity, a liquid will conform to the shape of its container.

2.2.1.2 Pressure

The pressure existing at any point in a liquid body at rest is caused by the atmospheric pressure exerted on the surface, plus the weight of the liquid above that point. This pressure is equal in all directions.

2.2.1.3 Temperature

For most liquids, an increase in temperature decreases viscosity, decreases specific gravity, and increases volume. Temperature affects:

- Type of pump construction
- Material selection
- Corrosive properties of a fluid
- Pumps flange pressure ratings

2.2.1.4 Head

The term "head" is used to represent the vertical height of a static column of liquid; it corresponds to the energy contained in the liquid per unit mass. Head can also be considered as the amount of work necessary to move a liquid from its original position to the required delivery position. In this case, the head includes the extra work necessary to overcome the resistance (friction) to flow. In general, a liquid at any point may have the following three types of head:

- *Static pressure head* represents the energy contained in the liquid due to its pressure.
- *Potential head* represents the energy contained in the liquid due to its position measured by the vertical height above some plane of reference.
- *Velocity head* represents the kinetic energy contained in the liquid due to its velocity.

The static discharge head of a pump is the height at which a pump can raise a fluid. If one disconnects the discharge pipe on a pump and extends it vertically the discharge head is the height at which the pump raises the fluid. As shown in Fig. 2.4, the more pressure the pump delivers the higher the head will be. Let us assume the head in Fig. 2.4 is 60 ft (18 m). As shown in Fig. 2.5, as the level in the suction tank (static

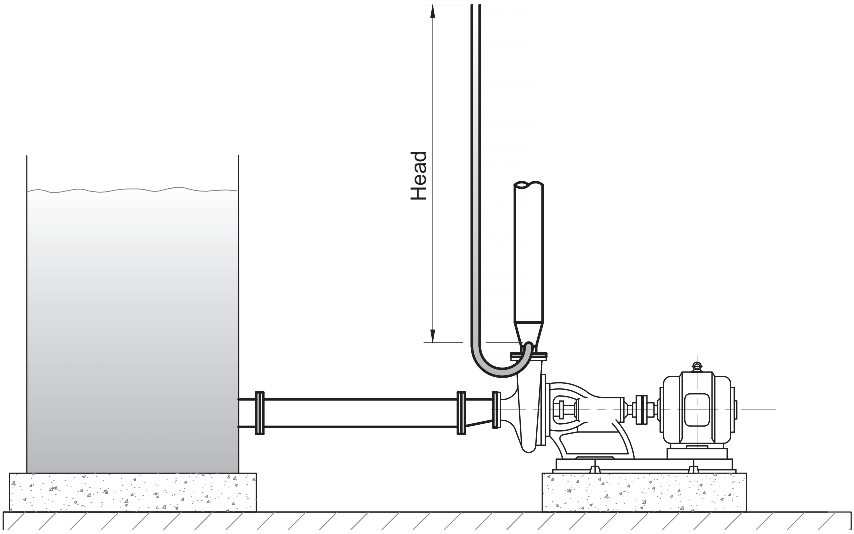


Fig. 2.4 The meaning of head.

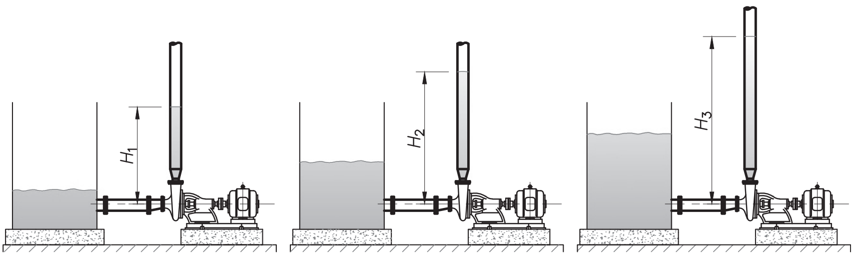


Fig. 2.5 How discharge head increases as the level in the suction tank increases.

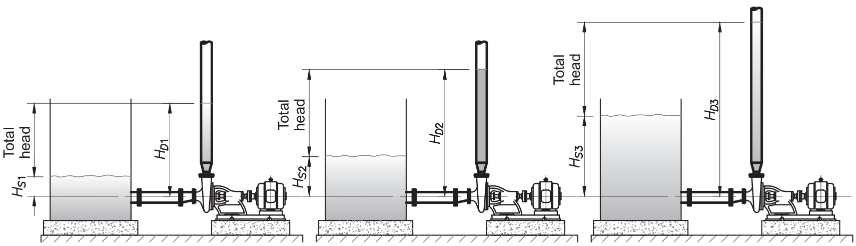


Fig. 2.6 The effects of increasing the suction head on discharge head and total head.

suction head) increases ($H_3 > H_2 > H_1$) the static discharge head increases and vice versa.

In order for a pump manufacturer to show the head capability for a specific pump, the suction head, H_S, is subtracted from the discharge head, H_D, to produce the total head, H_T, available. Fig. 2.6 shows the effects of increasing the suction head on discharge head and total head.

As shown in Fig. 2.7, if a pump is lifting a fluid from below the pump (suction lift) centerline the pump will still produce the same total head (H_T), but the discharge head (H_D) will be reduced.

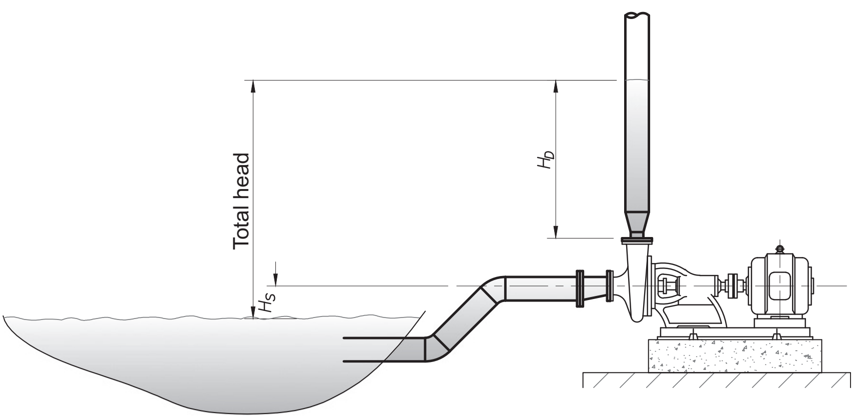


Fig. 2.7 Effect of low level on the pump suction (suction lift).

2.2.1.4.1 Centrifugal pumps

Operating pressure is expressed in feet (meters) of the fluid that is being pumped. Suction and discharge pressures are expressed as suction head and discharge head. Pressures are expressed in feet (meters) of head because it is more important to know how much a pump can raise the fluid it is pumping, rather than the amount of pressure the pump is adding to the fluid.

2.2.1.4.2 Positive displacement pumps

Operating pressures are almost always expressed in terms of pressure (psi)(bar).

The relationship of head to pressure is expressed as follows:

Field units

$$\text{Head (ft)} = \frac{\text{Pressure (psi)} \times 2.31}{\text{Specific Gravity of the Fluid}} \tag{2.1a}$$

SI units

$$\text{Head (m)} = \frac{\text{Pressure (kPa)} \times (0.102)}{\text{Specific Gravity of the Fluid}} \tag{2.1b}$$

Conversely,

Field units

$$\text{Pressure (psi)} = (SG)(h) \tag{2.2a}$$

SI units

$$\text{Pressure (kPa)} = (SG)(h) \tag{2.2b}$$

where

SG = density of the fluid
h = height of the fluid column above a reference point, ft (m)
Pressure = pressure, psi (kPa)

The head of fluid is not related to the area occupied by the fluid. The types of head are illustrated in Fig. 2.8. Water is the reference fluid. Density of fluids other than water can be determined from the following formula.

$$SG = \frac{\rho_f}{\rho_w} \tag{2.3}$$

where

SG = specific gravity of fluid
ρ_f = density of fluid, Ibs/ft^3, (kg/m^3)
ρ_w = density of water, Ibs/ft^3, (kg/m^3)
= 62.34 Ibs/ft^3 at 60°F
= 1000 kg/m^3 at 4°C

Since there are 144 in^2/ft^2, the following relationships exist.

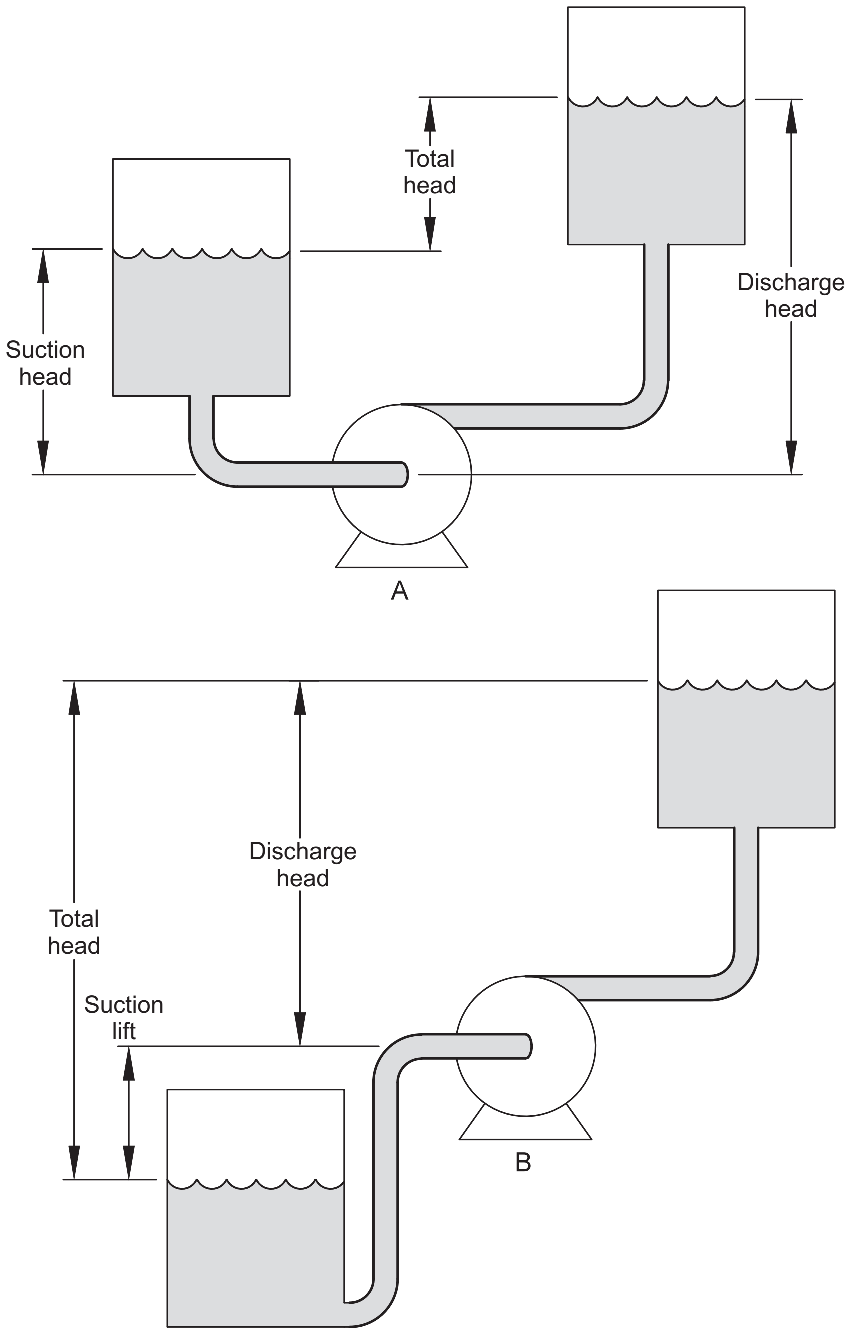


Fig. 2.8 Types of head. (A) Suction head. (B) Suction lift.

$$\text{Pressure (psig)} = \left(\frac{62.34\,\text{lbs}}{ff^3}\right) h(SG) \left(\frac{\text{ft}^2}{144\,\text{in}^2}\right) \tag{2.4}$$

$$= \frac{(SG)}{2.31} h \tag{2.5}$$

or

$$= 0.433h(SG) \tag{2.6}$$

Fig. 2.9 illustrates the relationship between head and pressure. As an example, 10 ft (3 m) of clean water (SG = 1.0) in a tank exerts a static pressure of 4.33 psi (29.41) kPa), while 100 ft (30 m) exerts 43.3 psi (294.11 kPa). In a kinetic energy type pump application, we would state 10 ft (3 m), while for the positive displacement pump, it would be correct to state 4.33 psi (29.41 kPa).

It is important to realize that although the heads of different fluids are the same, their pressures are different because of the difference in specific gravities. As an example, assume three 100-ft (30-m) tall tanks completely filled with gasoline, water,

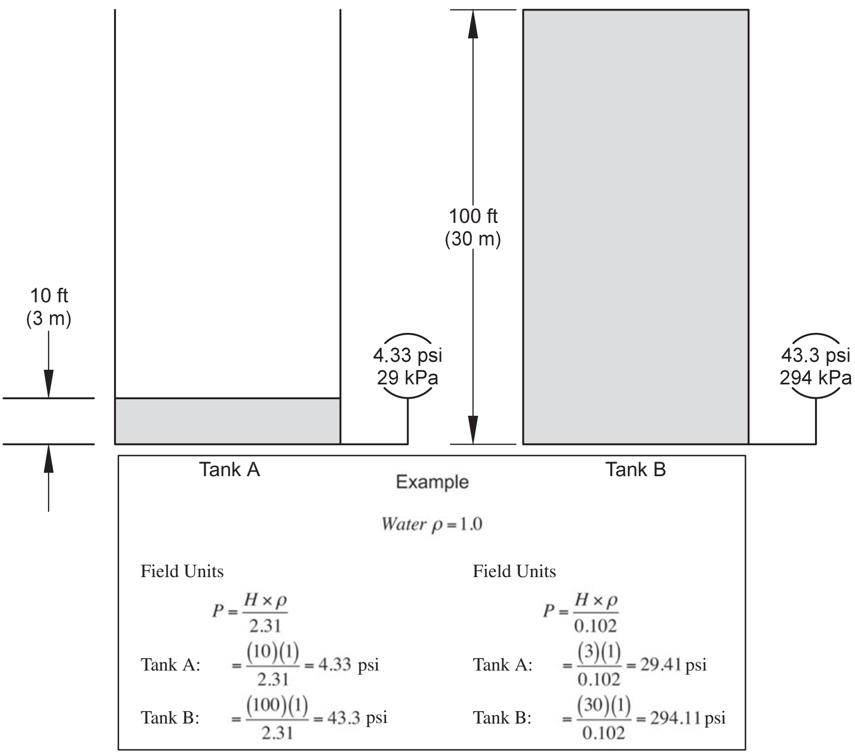


Fig. 2.9 Head vs pressure.

and molasses. The pressure measured at the bottom of each tank is different due to the differences of specific gravities of gasoline (0.75), water (1.0), and molasses (1.45).

2.2.1.5 Static suction head vs static suction lift

Static suction head is the vertical distance between a fluid level and a datum line when the supply is located above the datum (Fig. 2.10).

Static suction lift is the vertical distance between a fluid level and the datum line when the datum line is located above the supply (Fig. 2.11).

Datum line is the centerline of the pump inlet connection or the horizontal centerline of the first-stage impeller on a vertical pump.

Theoretical lift is the pressure to lift a fluid and comes from atmosphere pressure. At sea level, atmospheric pressure is 14.7 psia (100 kPa) (refer to Fig. 2.12). Theoretically, atmospheric pressure can lift a column of water 34 ft (10.2 m) if there are no air leakages (full vacuum in the suction line) into the system (refer to Fig. 2.13). This vertical distance (34 ft) (10.2 m) that a pump lifts water below the level of the pump is called theoretical lift.

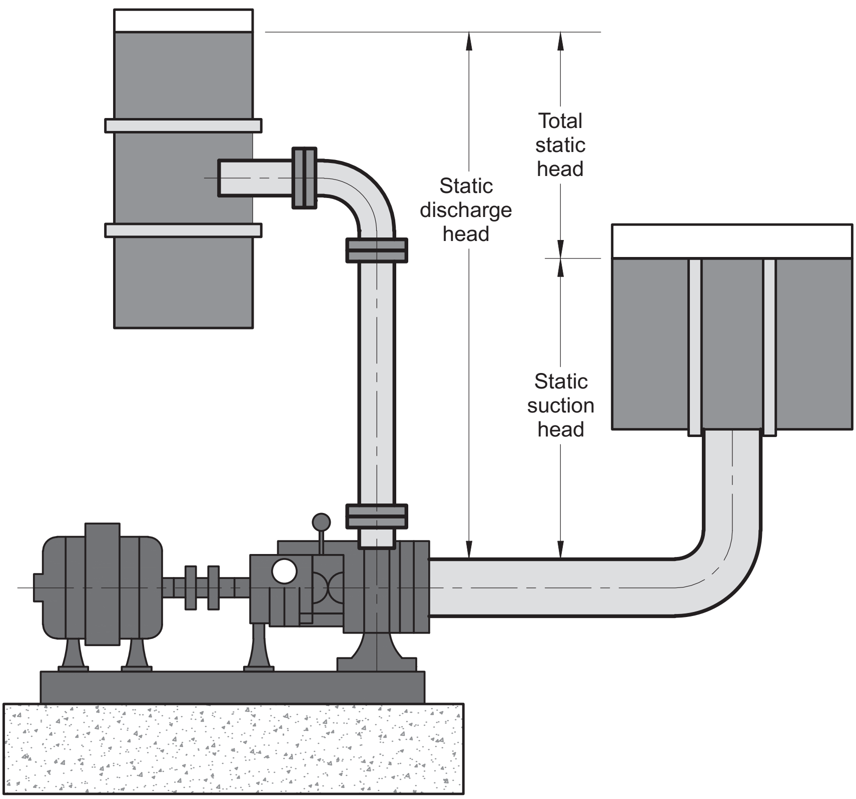


Fig. 2.10 Examples of static suction head.

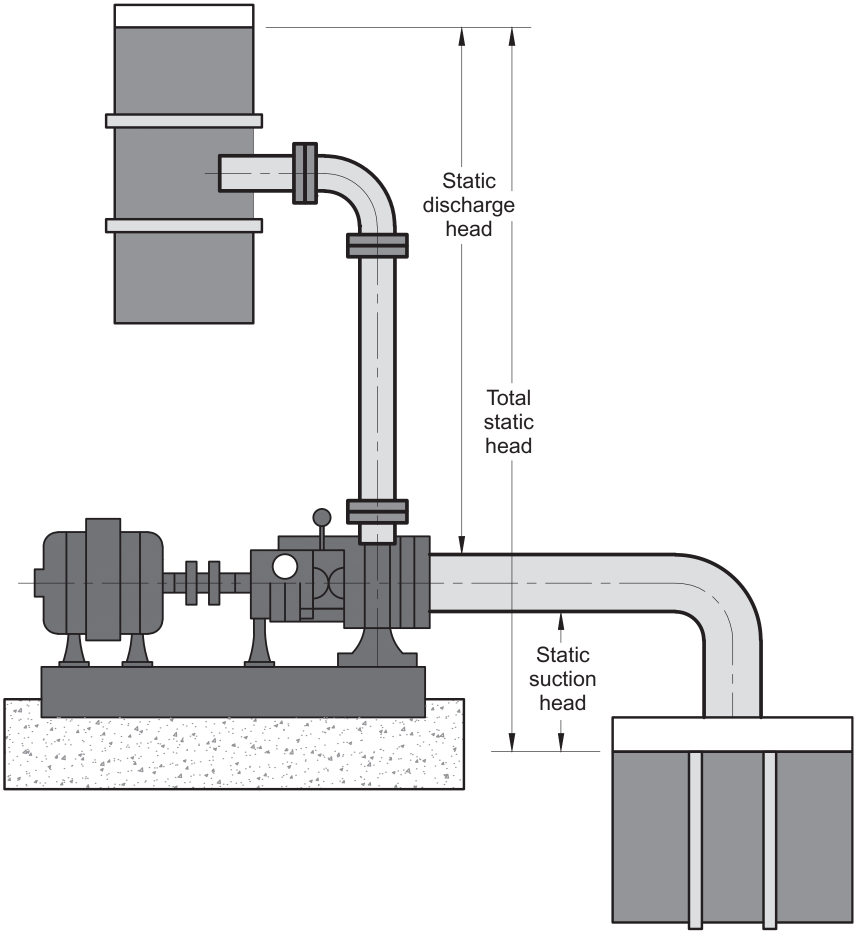


Fig. 2.11 Example of static suction lift.

Under perfect conditions.
Field units

$$\text{Pressure} = 0.433\,h\,(SG)$$

$$\text{Pressure} = 0.433h(SG) \tag{2.7}$$

$$14.7 = 0.433h(1.0)$$

Rearranging and solving for h

$$h = 34\,\text{ft}$$

SI units

$$h = 100\,\text{kPa}\,(0.102) = 10.2\,\text{m} \tag{2.1b}$$

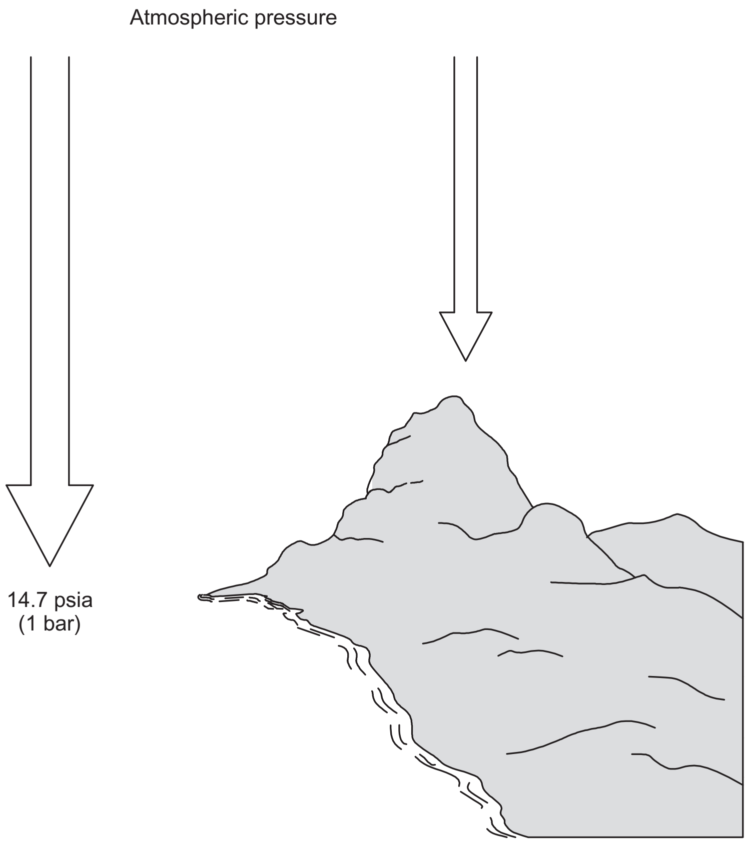


Fig. 2.12 Effects of elevation on atmospheric pressure.

Actual suction lift: Since a full vacuum is never achieved and since some lift is lost to friction in the suction line, the maximum actual suction lift for a positive displacement pump is about 22 ft (6.71 m) and 15 ft (4.5 m) for a centrifugal pump, when pumping water from an open air tank (Fig. 2.14). Positive displacement pumps can operate with lower suction pressures or higher suction lifts because they can create stronger vacuums. Suction lift will be greater if the pressure in a closed tank is greater than atmospheric pressure.

2.2.1.6 Submergence

Submergence is often confused with either suction static head or suction static lift. Submergence is defined differently for vertical and horizontal pumps. Fig. 2.15 illustrates the relationship between static head, static lift, and submergence.

Vertical pumps: Submergence relates the fluid level to the setting of the pump.

Horizontal pumps: Submergence relates to the amount of fluid level necessary in the source vessel or tank to prevent the formation of a vortex and the resulting drain of vapors into the pump.

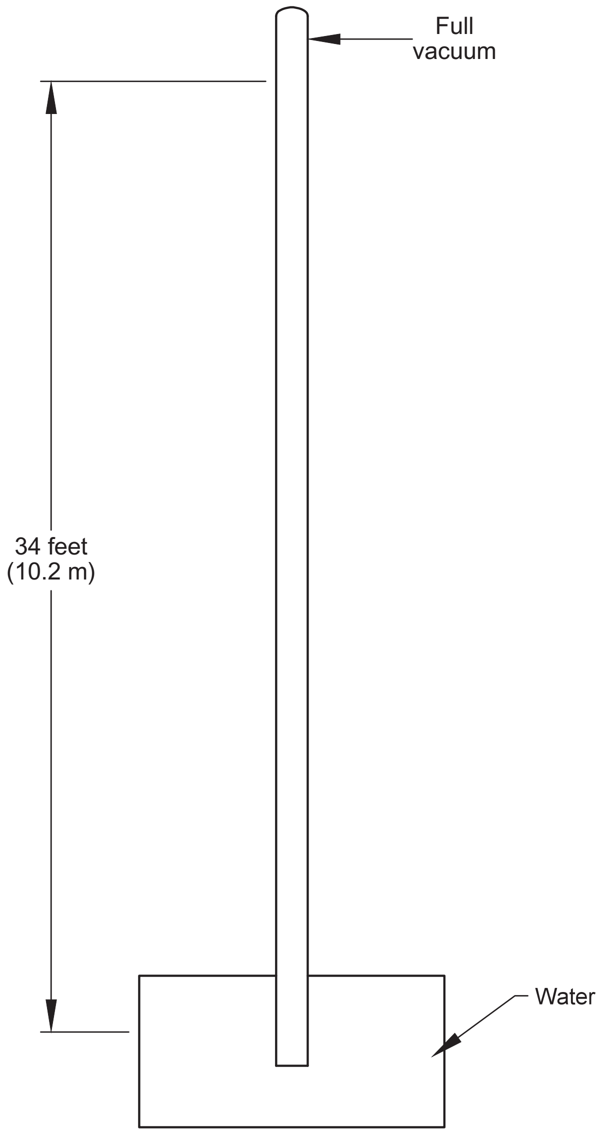


Fig. 2.13 Theoretical lift.

2.2.1.7 Vapor pressure

Vapor pressure is important when calculating the net positive suction head (NPSH). It is measured in psia (bara) and is a direct function of fluid temperature. At 60°F (15.6°C) the vapor pressure of water is 0.3 psia (0.7 ft). At 212°F (100°C) the vapor pressure of water is 14.7 psia (101 kPa).

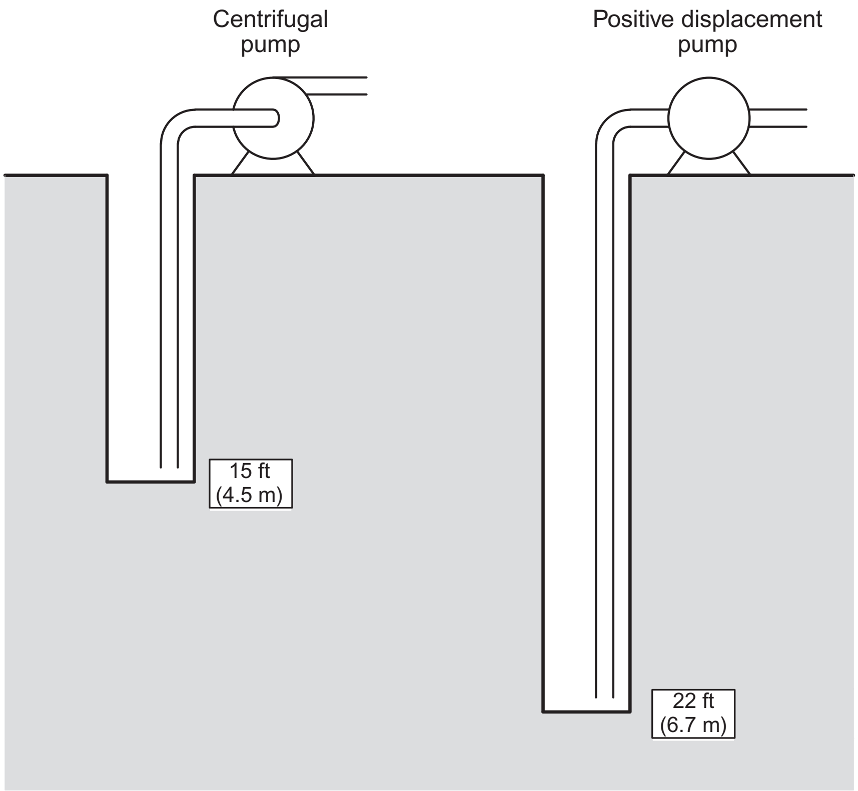


Fig. 2.14 Actual suction lift for a centrifugal and positive displacement pump.

2.2.1.8 Suspended solids

The amount and type of suspended solids entrained in the fluid can affect the characteristics and behavior of that fluid. Increased concentrations of solids increase:

- Specific gravity
- Viscosity
- Abrasiveness

The type and concentration of suspended solids can affect

- Style of pump selected
- Materials of construction

Suspended solids affect the selection of impeller design in centrifugal pumps, which in turn affects:

- Wear rate
- Efficiency
- Power consumption

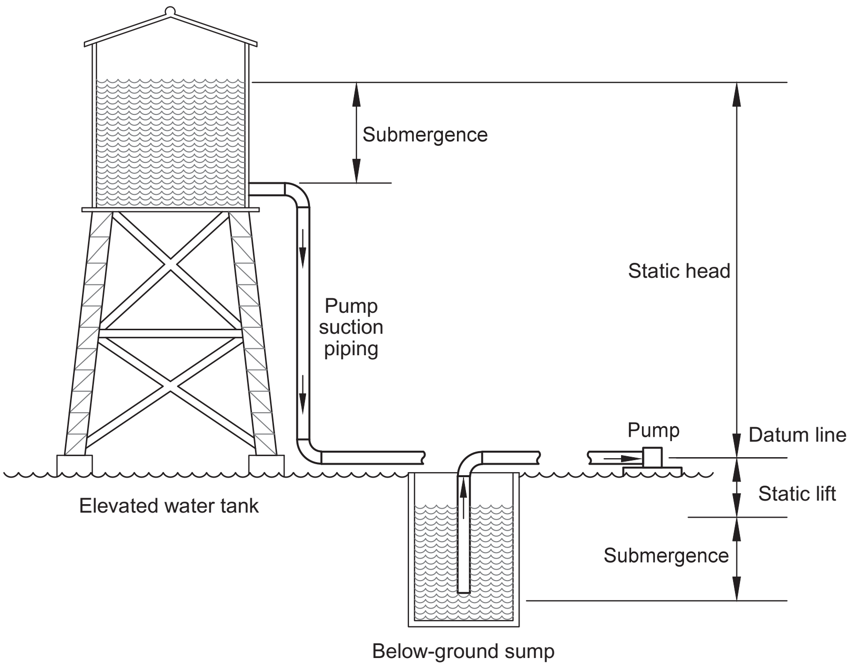


Fig. 2.15 Static head, static lift, and submergence in hydrostatics.

2.2.1.9 Dissolved gases

Small amounts of dissolved gases have little effect on flow rate or other pumping requirements. If large amounts of gas enter the liquid through piping leaks or as a result of a vortex in vessels, the specific gravity of the fluid will decrease and thus will also reduce the amount of NPSH available at the pump suction.

2.2.1.10 Viscosity

Viscosity offers resistance to flow due to friction within the fluid. Viscosity levels have a significant impact on

- Pump type selection
- Efficiency
- Head capability
- Warm-up

High viscosity affects centrifugal pumps by

- Decreasing the pump efficiency
- Decreasing the head performance
- Increasing the power requirements

Viscosity of all fluids varies with temperature. Reciprocating pumps can operate with viscosities up to 660 cSt (3000 Saybolt Second Universal (SSU)) and rotary pumps up

Table 2.1 Typical pump applications as a function of viscosity

Viscosity SSU (centistokes)	Example fluids @ 100°F (38°C)	Guidelines
<50 (7.4)	Water Beer Gasoline Alcohol 0.85 crude oil	Minimum for rotaries, centrifugals efficiency
< 150 (32.1)	0.90 crude oil Tri-ethylene glycol	Centrifugal always preferred over rotaries where a choice is available
150 (32.1)	0.95 crude oil	Centrifugal head-capacity curve begins to deteriorate
150–1000 (32.1–220)	SAE crankcase oils	Centrifugal still preferred over rotaries where choice is available despite performance falloffs
1000–3000 (220–660)	First cut molasses	Centrifugal seldom used, reciprocating pumps generally used
> 3000 (660)	Blackstrap molasses Asphalts	Centrifugal should never be used

to 6,000,000 SSU (1,320,000 cSt (6,000,000 SSU). A Saybolt viscometer measures the number of seconds it takes for a liquid to flow through a specified orifice and it is expressed as Saybolt Seconds Universal (SSU). Table 2.1 provides guidelines on typical practices.

2.2.1.11 Corrosive nature

The corrosive nature of the fluid being pumped has a bearing on pump type selection, materials of construction, and corrosion allowance. Special mechanical seals and flushing arrangements may be required.

2.2.2 Hydrodynamics

2.2.2.1 Overview

Hydrodynamics is the study of fluids in motion.

2.2.2.2 Continuity equation

The continuity equation (conservation of mass) states:

$$(\rho \times A \times V)_1 = (\rho \times A \times V)_2 \tag{2.8}$$

where

V = Velocity, ft/s (m/s)

Subscripts 1 and 2 refer to the continuity of flow that exists between two different points in the system. For incompressible fluids this simplifies to

$$(A \times V)_1 = (A \times V)_2 \tag{2.9}$$

2.2.2.3 Bernoulli's equation

Bernoulli's Law states that as a fluid flows from one point to another in a piping system the total of potential, static and velocity head at the upstream point (subscript 1) equals the total of the three heads at the downstream point (subscript 2) plus the friction drop between points 1 and 2. Bernoulli's equation states the following:

$$\frac{V_1^2}{2g} + \frac{P_1}{\rho} + Z_1 = \frac{V_2^2}{2g} + \frac{P_2}{\rho} + Z_2 + h_f \tag{2.10}$$

where

$\frac{V^2}{2g}$ = velocity head, ft/s (m/s)
P/ρ = pressure head, ft (m)
Z = elevation head, ft (m)
h_f = head losses, ft (m)

An examination of each of the terms provides a better understanding of the general equation for modeling a pumping system, which is presented later in this chapter.

2.2.2.4 Velocity head

Potential energy that has been converted to kinetic energy is expressed by the following term:

$$\frac{V^2}{2g} \tag{2.11}$$

The velocity head is the first term in the momentum equation
where

V = Average velocity of the liquid in the pipe, ft/s (m/s)
= 0.4085 (gpm/d^2)
g = Acceleration due to gravity, ft/s^2 (Table 2.2)
= 32.17 ft/s^2 (9.81 m/s^2 at sea level at 45° north latitude
= Values vary with latitude and altitude

Velocity head is often expressed as
Field units

$$= 0.00259 \frac{(\text{gpm})^2}{d^4} \tag{2.12}$$

Table 2.2 Relation of acceleration of gravity (g) to latitude

Latitude degree	**Acceleration due to gravity**[a] **(ft/s^2)**
0	32.088
10	32.093
20	32.108
30	32.130
40	32.158
45	32.174
50	32.187
60	32.215
70	32.238
80	32.253
90	32.258

[a]At sea level; for altitudes above sea level, subtract 0.003 ft/s^2 for each 1000 ft.

or

$$= 0.0155\ V^2 \tag{2.13}$$

where

gpm = gallons per minute flow rate
d = inside pipe diameter, in (cm)

Velocity heads increase the amount of work required of a pump. It is usually not included in actual system calculations when piping velocities are kept within the prescribed limits recommended in *Volume 3* "*Plant Piping and Pipeline Systems*." Velocity heads are included in the total dynamic head on centrifugal pump curves.

2.2.2.5 Static head

The static head corresponding to any pressure depends on the weight of the liquid according to the following equation:
Field units

$$\text{Head (ft)} = \frac{\text{Pressure (psi)} \times (2.31)}{SG} \tag{2.14a}$$

SI units

$$\text{Head (m)} = \frac{\text{Pressure (kPa)}(0.102)}{SG} \tag{2.14b}$$

An examination of Eqs. (2.14a), (2.14b) shows that head is a constant for a pump at a specific operating condition; however, the differential pressure developed is a direct function of the liquid being pumped. For example, if the head is 100 ft and the specific gravity is equal to 1.0, then differential pressure increase is equal to $\frac{[(100)(1.0)]}{(2.31)} = 43.3$ psi but if SG = 0.8, then the differential pressure increase

$= \frac{[(100)\times(0.80)]}{2.31} = 34.6$psi. This fact is often overlooked by inexperienced pump designers and is good to keep in mind.

The velocity head energy component is often used in system head calculations as a basis for establishing entrance losses, losses in valves and fittings, losses in other sudden enlargements, and exit losses by applying the appropriate resistance coefficient (K) to the $V^2/2\,g$ term. In system head calculations for high head pumps, the change in velocity head is a small percentage of the total head and is not significant. In low head pumps, however, it can be a substantial percentage and must be considered.

2.2.2.6 Head losses (h_f)

Head loss is potential energy that is converted to kinetic energy. Head losses are due to the frictional resistance of the piping system (pipe, valves, fittings, entrance, and exit losses). Unlike velocity head, friction head cannot be ignored in system calculations. Values vary as the square of the flow rate. Head losses can be a significant portion of the total head. The importance of head losses depends upon the magnitude of the total head. For example, an error of 6 ft (1.8 m) in estimating 2600 ft (780 m) of head is insignificant while an error of 6 ft (2 m) in estimating 30 ft (9 m) of head for a crude oil transfer pump could be significant.

2.2.2.7 Control losses

Control losses occur on the discharge side of a centrifugal pump that has been equipped with a flow control (backpressure) valve for the purpose of controlling flow rate. Again, the potential energy is converted to kinetic energy. Next to static head, control losses are frequently the single most important factor in calculating the pump's total dynamic head. For pump applications, control losses are treated separately from head losses, even though they are included in the "h_f" term in the momentum equation.

2.2.2.8 Acceleration head (H_A)

Acceleration head is used to describe the losses associated with the pulsating flow associated with reciprocating pumps. It is included in the "h_f" term of the momentum equation.

2.2.2.9 Suction head (H_S)

The suction head is the sum of the suction vessel operating gauge pressure (converted to feet (m)), the vertical distance between the suction vessel fluid level and the pump reference points, less head losses in the suction piping (discounting velocity head). Expressed as

Field units

$$H_s = \frac{P_1 \times 2.31}{SG} + H_1 - \frac{P_{f1} \times 2.31}{SG} \tag{2.15a}$$

SI units

$$H_s = \frac{P_1 \times (0.102)}{SG} + H_1 - \frac{P_{f1} \times (0.102)}{SG} \tag{2.15b}$$

Reducing

Field units

$$= \frac{(P_1 - P_{f1})(2.31)}{SG} + H_1 \tag{2.16a}$$

SI units

$$= \frac{(P_1 - P_{f1})(0.102)}{SG} + H_1 \tag{2.16b}$$

where

H_s = suction head of fluid being pumped, ft (m)
P_1 = Suction vessel operating pressure, psig (kPa)
= Lowest normal operating pressure on the surface of the pumped fluid
H_1 = Height of fluid in suction vessel above pump reference point, ft (m)
= The vertical distance, in feet (m), between the surface of the pumped liquid and the center of a horizontal pump suction flange
= Value is negative (−) when the pump is above the liquid surface.
P_{f1} = Pressure drop, line losses, due to friction in the suction piping, psi (kPa)

2.2.2.10 Discharge head (H_D)

The discharge head is the sum of the discharge vessel operating gauge pressure (converted to feet or meters), the liquid level in the discharge vessel above the pump reference point, pressure drop due to friction in the discharge piping, and control losses (discounting velocity head). Expressed as

Field units

$$H_D = \frac{P_2 \times 2.31}{SG} + H_2 + \frac{P_{f2} \times 2.31}{SG} + \frac{P_c \times 2.31}{SG} \tag{2.17a}$$

SI units

$$H_D = \frac{P_2 \times (0.102)}{SG} + H_2 + \frac{P_{f2} \times (0.102)}{SG} + \frac{P_c \times (0.102)}{SG} \tag{2.17b}$$

Reducing

Field units

$$= \frac{(P_2 + P_{f2} + P_c)(2.31)}{SG} + H_2 \tag{2.18a}$$

SI units

$$= \frac{(P_2 + P_{f2} + P_c)(0.102)}{SG} + H_2 \tag{2.18b}$$

where

H_D = discharged head of fluid being pumped, ft (m)
P_2 = discharge vessel operating pressure, psig (kPa)
H_2 = operating or normal height of fluid in the discharge vessel above the pump reference point, ft (m)
P_{f2} = pressure drop, line losses, due to friction in the discharge piping, psi (kPa)
P_c = discharge flow control valve losses, ft (m)

2.2.2.11 Total dynamic head (TDH)

The total dynamic head (TDH) is the difference between the pumping system discharge and suction heads. It is equal to the difference in pressure gauge readings (converted to feet or meters) across and existing operating pump (discounting velocity heads).

A pump's total dynamic head (TDH) is the difference between the suction and discharge heads (refer to Fig. 2.16)

$$\text{TDH} = H_D - H_S \tag{2.19}$$

Substituting
Field units

$$= \frac{(P_2 - P_1 + P_{f1} + P_{f2} + P_c)(2.31)}{SG} + H_2 - H_1 \tag{2.20a}$$

SI units

$$= \frac{(P_2 - P_1 + P_{f1} + P_{f2} + P_c)(0.102)}{SG} + H_2 - H_1 \tag{2.20b}$$

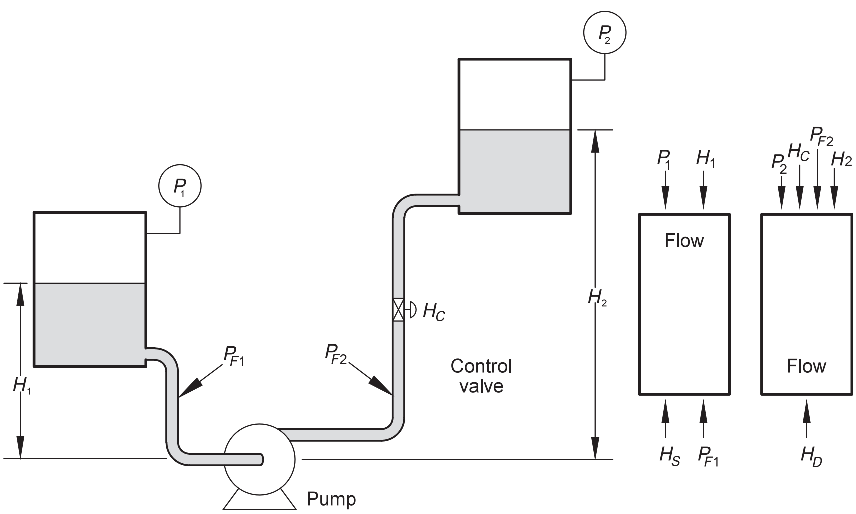


Fig. 2.16 Total head calculations.

where

TDH = total dynamic head, ft (m)

A pump's required discharge pressure rating is the combination of the maximum expected suction pressure and the maximum pressure developed by the pump. Centrifugal pumps require suction head and total dynamic head values be converted to units of pressure whereas reciprocating pumps do not require conversions.

2.2.2.12 Net positive suction head (NPSH)

The NPSH is defined as the total suction head (H_S), in feet (m) of liquid (absolute at the pump centerline or impeller eye) less the vapor pressure (in feet (m)), of the liquid being pumped.

2.2.2.13 Net positive suction head required ($NPSH_R$)

The $NPSH_R$ is the amount of NPSH required to move and accelerate the fluid from the pump suction into the pump itself (refer to Fig. 2.17). It is determined either by test or calculation by the pump manufacturer for the specific pump under consideration. The $NPSH_R$ is a function of

- Fluid geometry
- Smoothness of the surface areas

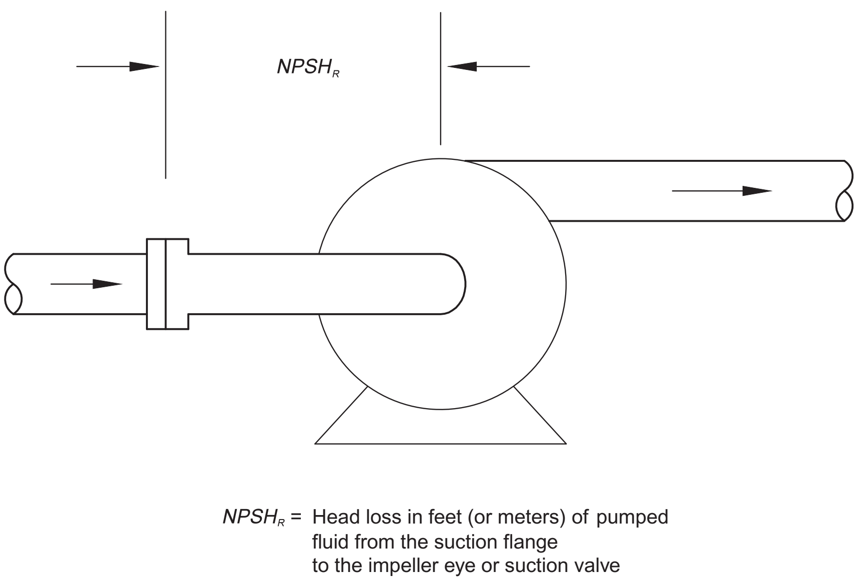


Fig. 2.17 Net positive suction head required ($NPSH_R$).

For centrifugal pumps, other factors included are as follows:

- Type of impeller
- Design of impeller eye
- Rotational speeds

2.2.2.13.1 $NPSH_R$ reductions for centrifugal pumps

The $NPSH_R$ requirement for centrifugal pumps is determined on the basis of handling cool water. Field experience and laboratory testing have confirmed that centrifugal pumps handling gas-free hydrocarbon fluids and water at elevated temperatures will operate satisfactorily with harmless cavitation and less $NPSH_A$ than would be required for cold water.

Fig. 2.18 shows the reductions in NPSH that should be considered when handling hot water and certain gas-free pure hydrocarbon liquids. The use of Fig. 2.18 is subject to certain limitations some of which are summarized as follows:

(1) The NPSH reductions are based on laboratory test data at steady-state suction conditions and on gas-free pure hydrocarbon liquids shown; its application to other liquids must be considered experimental and is not recommended.

(2) No NPSH reduction should exceed 50% of the NPSH required for cold water or 10 feet (3 m) whichever is smaller.

(3) In the absence of test data demonstrating NPSH reductions >10 ft (3 m) the graph has been limited to that extent and extrapolation beyond that point is not recommended.

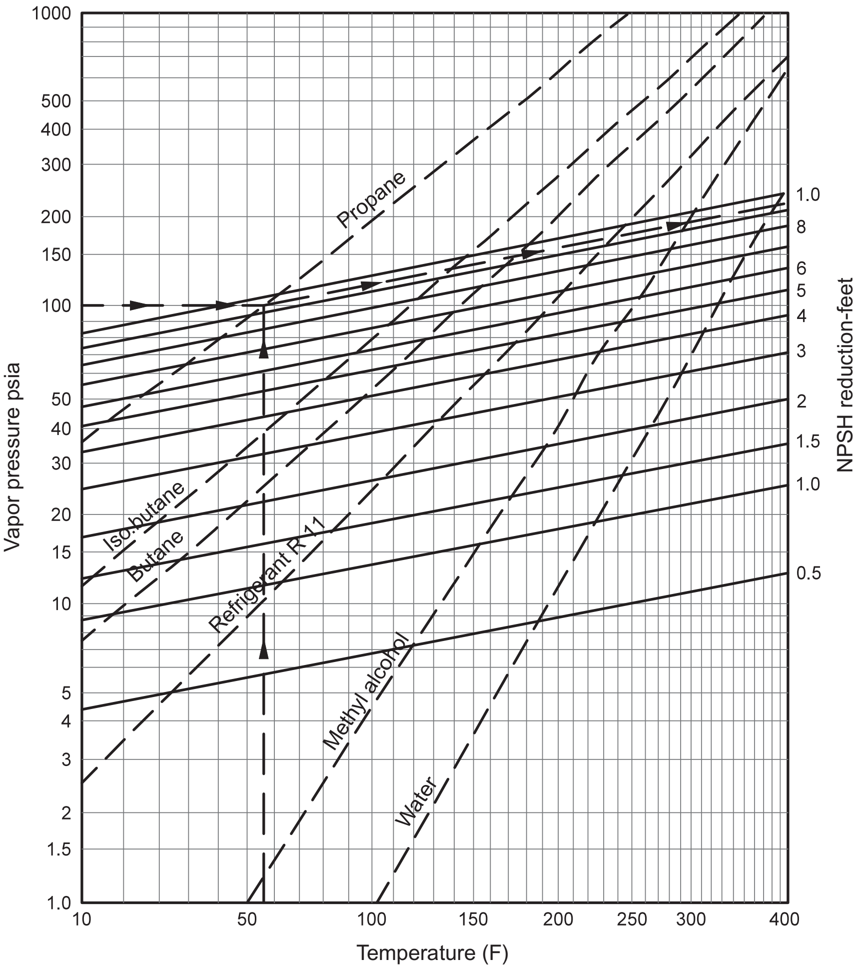
(4) Vapor pressure for the liquid should be determined by the bubble point method—do not use the Reid vapor pressure.

(5) Do not use the graph for liquids having entrained air or other noncondensable gases which may be released as the absolute pressure is lowered at the entrance to the impeller, in which case additional NPSH may be required for satisfactory operation.

(6) In the use of the graph for high-temperature liquids, particularly with water, due consideration must be given to the susceptibility of the suction system to transient changes in temperature and absolute pressure which might require additional NPSH to provide a margin of safety, far exceeding the reduction otherwise permitted for steady-state operation.

The procedure in using Fig. 2.18 is best illustrated by going through an example. Assume a pump requires a $NPSH_R$ of 16 ft (4.88 m), based on cold water at the design capacity, is required to handle pure propane at 55°F (12.8°C) which has a vapor pressure of approximately 100 psia (689.5 kPa). Entering Fig. 2.18 at a temperature of 55°F (12.8°C) and intersecting the propane curve shows a reduction of 9.5 ft (2.9 m) (which is greater than one half the cold water $NPSH_R$). The corrected value of the $NPSH_R$ is one half the cold water $NPSH_R$ or 8 ft (2.4 m).

Assume the same pump has another application to handle propane at 14°F (−10°C) where its vapor pressure is 50 psia (344.7 kPa). In this case the graph in Fig. 2.16 shows a reduction of 6 ft (1.8 m) which is less than one-half of the cold water NPSH. The corrected value of NPSH is, therefore, 16 ft (4.9 m) less 6 ft (1.8 m) or 10 ft (3 m). For a more detailed discussion on the use of this chart and its limitations, one should follow the requirements of the *Hydraulic Institute* Standards.



Note: Note chart has been constructed from test data obtained using the liquids shown. For applicability to other liquids refer to the text.

Fig. 2.18 NPSH reduction for pumps handling hydrocarbon liquids and high-temperature water.
Courtesy of Ingersoll-Dresser Pumps.

*2.2.2.14 Net positive suction head available (*NPSH$_A$*)*

The $NPSH_A$ is a critical factor in pump performance. It is a result of the suction system design. In practice, $NPSH_A$ is the differential pressure between (1) the actual pressure at the lowest pressure point in the pump, and (2) the pressure at which the liquid begins to vaporize (flash). $NPSH_A$ is the "available" pressure above the liquid's vapor pressure that prevents vaporization (or cavitation). As the liquid accelerates into the spinning impeller eye, its pressure drops. If pressure falls below the vapor pressure cavitation will result. For centrifugal pumps, the $NPSH_A$ is always expressed in feet of the liquid being pumped.

A pump's $NPSH_A$ is usually set by the height of the lowest liquid level in a column, vessel, or tank. With boiling liquids, static head (H_1) is the only source of $NPSH_A$. (Elevating the vessel or column, or establishing a low liquid level (LSL) which satisfies pump NPSH requirements is preferred over the use of pumps in a pit or vertical in a suction can.)

The $NPSH_A$ must be equal to or greater than $NPSH_R$ in order not to cavitate or flash across the pump. Cavitation results in decreased efficiency, capacity, and head. Flashing results as the pump cavity is filled with vapors, and as a result, the pump becomes vapor locked. The end result of this operation is usually pump seizure.

$NPSH_A$ is not a function of the pump itself but of the piping system for the pump. It can be calculated from the following (refer to Fig. 2.19)

Field units

$$NPSH_A = \frac{\left(P_1 + P_a - P_V - P_{f1}\right)(2.31)}{SG} + H_1 \tag{2.21a}$$

SI units

$$NPSH_A = \frac{\left(P_1 + P_a + P_V + P_{f1}\right)(0.102)}{SG} \tag{2.21b}$$

where

P_a = atmospheric pressure, psia (kPa absolute)
P_v = liquid vapor pressure at pumping temperature, psia (kPa absolute)

A primary factor in calculating the $NPSH_A$ for a pump is the vapor pressure of the liquid handled. One commonly used method, Reid vapor pressure, requires a certain amount of liquid to be evaporated in the measuring apparatus before the vapor pressure is indicated. Such vapor pressures are too low for determining when gas evolution will start (the point that will affect pump performance). This error is variable, being small for fractioned liquids and greater for crudes. The true vapor pressure (TVP) at the pumping temperature should be used for $NPSH_A$ calculations rather than vapor pressure by the Reid method.

When determining the TVP, one should not overlook the possibility of dissolved gases in the liquid. A frequent cause of NPSH trouble is dissolved or entrained gas in the liquid pumped. When tested by the bubble point method, water which has been aerated has a higher "vapor pressure" than water which has not been aerated. The same is true for hydrocarbons or other liquids. When the pressure of a liquid containing dissolved gases is reduced, the gas dissolved in the liquid may evolve and cause an effect similar to cavitation.

Vapor pressure is a function of temperature alone for any given composition of liquid. For some fluids, a small increase in temperature causes a relatively large increase in vapor pressure. When selecting a pump for such a fluid (e.g., water), the $NPSH_A$ should be calculated at the highest probable fluid temperature.

Fig. 2.20 is a vapor pressure curve for water. For other fluids, refer to the GPSA Engineering Data Book, Hydraulic Institute Engineering data Book or similar. The

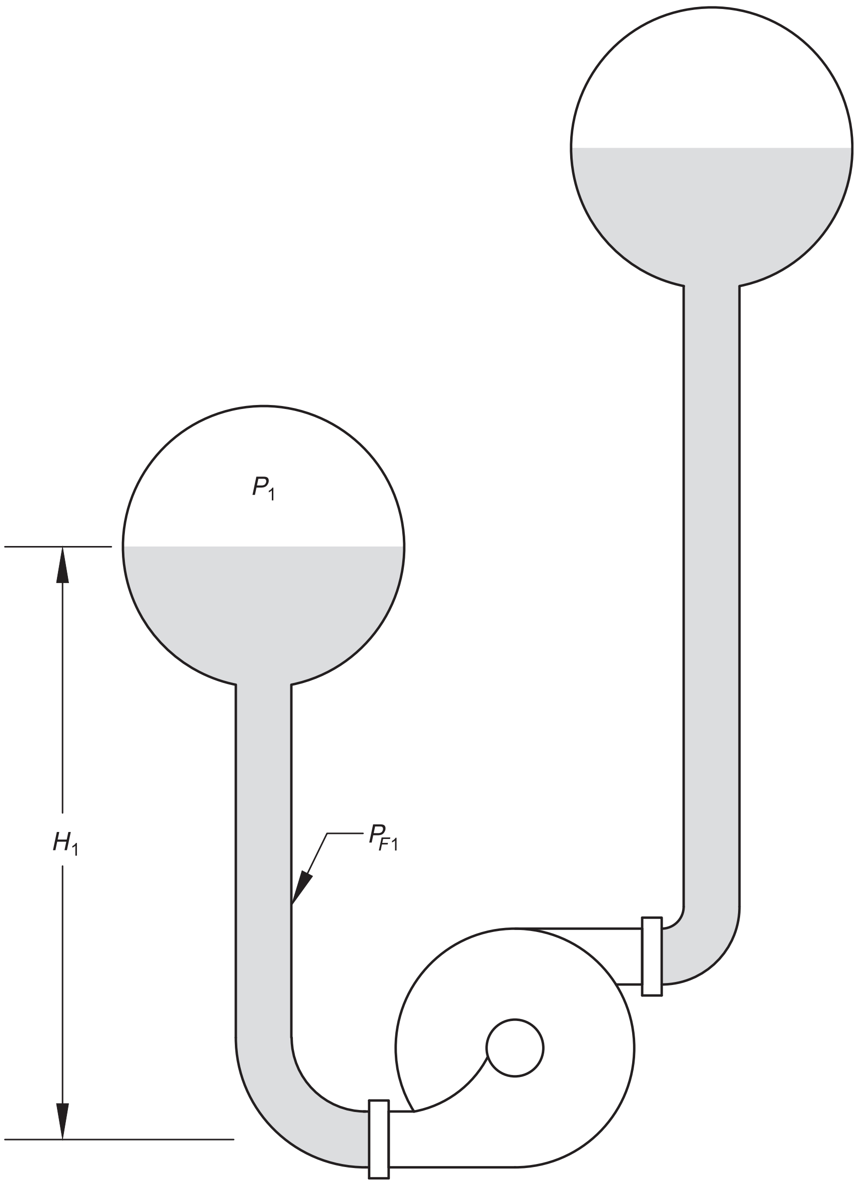


Fig. 2.19 Net positive suction head available $NPSH_A$.

$NPSH_A$ decreases linearly with increases in fluid temperatures and pipe friction losses. Since pipe friction losses vary as the square of the flow, $NPSH_A$ also varies as the square of the flow. Thus, $NPSH_A$ will be the lowest at the maximum flow requirement. Accordingly, it is important to recognize the need for calculating $NPSH_A$ (and $NPSH_R$) at maximum flow conditions as well as maximum fluid temperature, not just at design.

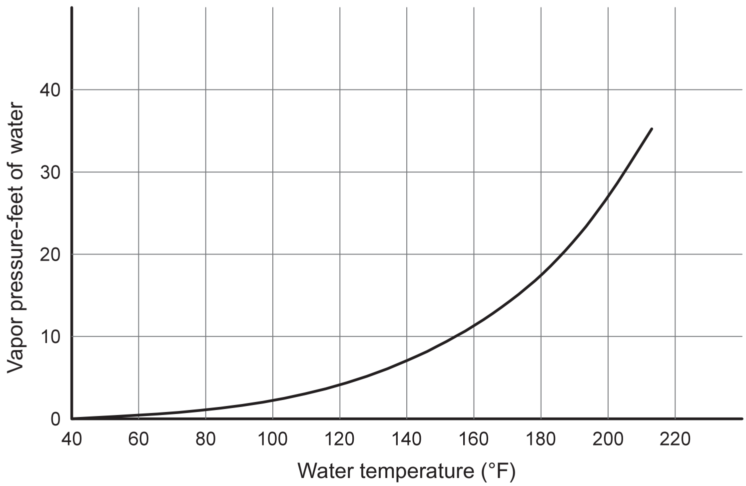


Fig. 2.20 Vapor pressure of water.

Unless subcooled, a pure component hydrocarbon liquid is typically in equilibrium with the vapors in a pressure vessel. Thus, increases in the vessel operating pressure are almost fully offset by a corresponding increase in the vapor pressure. When this happens

Field units

$$NPSH_A = H_1 - P_{f1}\left(\frac{2.31}{SG}\right) \tag{2.22a}$$

SI units

$$NPSH_A = H_1 - P_{f1}\left(\frac{0.102}{SG}\right) \tag{2.22b}$$

2.2.2.14.1 Acceleration head considerations for reciprocating pumps

The $NPSH_A$ for a reciprocating pump is calculated in the same manner as for a centrifugal pump, except in the $NPSH_R$ for a reciprocating pump some additional allowances must be made for the reciprocating action of the pump; this additional requirement is termed *acceleration head*. This head is required to accelerate the liquid column on each suction stroke so that there will be no separation of this column in the pump or suction line.

If this minimum condition is not met the pump will experience a fluid knock caused when the liquid column, which has a vapor space between it and the plunger (piston), overtakes the receding plunger (piston). This knock occurs approximately two-thirds of the way through the suction stroke. If sufficient acceleration is provided for the liquid to completely follow the motion of the receding face of the plunger (piston), this knock will disappear.

If there is insufficient head to meet minimum acceleration requirements of NPSH, the pump will experience cavitation, resulting in loss of volumetric efficiency.

In addition, serious damage can occur to the plungers (pistons), valves, and packing due to the forces released in collapsing the gas or vapor bubbles.

The head required to accelerate the fluid column is a function of the length of the suction line, the average velocity in this line, the rotating speed, the type of pump, and the relative elasticity of the fluid and the pipe, calculated as follows:

$$H_A = (LVnC)/Kg \tag{2.23}$$

where

H_A = acceleration head, feet (m)
L = length of suction line, feet (m)
V = velocity in suction line, ft/s (m/s)
  = [(gpm)(0.321)]/[area of the suction pipe, in^2]
n = pump speed, rpm
C = 0.200 for duplex, single-acting
  = 0.115 for duplex, double-acting
  = 0.066 for triplex, single or double-acting
  = 0.040 for quintuplex, single or double-acting
  = 0.028 for septuplex, single or double-acting
  = 0.022 for nonuplex, single or double-acting

Note: The constant "C" will vary from the earlier values for unusual ratios of connecting rod length to crank radius.

K = a factor representing the reciprocal of the fraction of the theoretical acceleration head which must be provided to avoid a noticeable disturbance in the suction line (K = 2.5 for hot oil; 2.0 most hydrocarbons; 1.5 for amine, glycol, water; 1.4 for deaerated water; and 1.0 for liquids with small amounts of entrained gases)
g = gravitation constant
= 32.174, ft/s^2 (9.81 m/s^2 at sea level at 45° north latitude)

The suction piping usually consists of pipes of various sizes. The acceleration head should be calculated for each section separately. The total acceleration head is obtained by adding the acceleration heads for each section.

2.2.2.15 NPSH margin

The NPSH margin is the difference between $NPSH_A$ and $NPSH_R$. The Hydraulic Institute recommends 3 to 5 ft (1 to 1.5 m) for centrifugal pumps and 7 to 10 ft (2 to 3 m) for reciprocating pumps. Perhaps a better approach is to specify a pump with a $NPSH_A$ that is less than what was calculated. We can use the following relationship

$$\text{Specified } NPSH_A = \frac{\text{Calculated } NPSH_A}{\text{Safety Factor}} \tag{2.24}$$

Typical safety factors

1.00 = existing system
1.10 = new services that have stable and well-controlled suction conditions

1.25 = new or old services that tend to have rapid, frequent, or severe fluctuations in suction conditions, such as a boiler feed water pump

Availability of pumps and pump models depends upon the calculated value of $NPSH_A$. For example

$NPSH_A < 7$ ft (2.25 m): Only a small number of pumps could be selected.
$NPSH_A > 25$ ft (7.5 m): Virtually any pump can be considered.

2.2.2.16 Inadequate suction conditions

When a new system offers insufficient NPSH margin for optimum pump selection, either

- $NPSH_A$ must be increased,
- $NPSH_R$ must be decreased, or
- Both

To increase the $NPSH_A$ the following can be done:

- Increase the static head (H_S)
    - Raise/add liquid level from reservoir, expansion tank, and so on
    - Lower the elevation of the selected pump
    - Reduce suction piping friction losses
    - Add a booster pump
    - Increase pressure in the system
- Add a pulsation damper
- Cool the liquid

A properly installed pulsation damper with a short, full-size connection to the pump or suction pipe can absorb the cyclical flow variation and reduce the pressure fluctuation in the suction pipe to that corresponding to a length of 5 to 15 (1.5 to 4.5 m) pipe diameters, if kept properly charged.

There is a similar pressure fluctuation on the discharge side of every power pump, but it cannot be analyzed as readily because of the greater influence of liquid and piping elasticity and the smaller diameter and much greater length of the discharge line in most applications. However, a pulsation damper can be just as effective in absorbing the flow variation on the discharge side of the pump as on the suction side and should be used if pressure fluctuation and piping vibration is a problem.

There are several ways to correct NPSH problems, depending on whether the system is an open or closed system. In an open system, one can raise the liquid level to increase suction pressure or lower the pump. Sometimes one may need to increase the suction line size in order to lower the friction losses. A closed system may require installation of a booster pump, raising system pressure, or maybe a different pump with a lower $NPSH_R$ requirement.

To reduce $NPSH_R$ the following can be done:

- Use slower speeds
- Use double-suction impeller

- Use large impeller eye
- Change to a low $NPSH_R$ pump
- Oversize pump
- Use different design impellers or inducers
- Use several smaller pumps with lower $NPSH_R$'s in parallel
- Use an inducer.

Inducers have severely restricted operating flow range abilities relative to noninducer pumps. Inducers should never be used in erosive services.

Some of the corrections suggested before are impractical and could instead simply require a change of pumps.

When an existing pumping system exhibits insufficient NPSH margin, it is too late to use the aforementioned solutions without going through an expensive change. The majority of these problems can be traced to suction line flow restrictions (orifice plates, plugged strainers, partially closed valves, etc.) and inadequate source tank fluid levels (including vortexing).

Example 2.1. Calculation of acceleration head (field units)
Given:

(a) A 2″ x 5″ triplex pump running at 360 rpm displaces 80 gpm of water.
(b) Suction line consists of 4 ft of 4-in. pipe and 20 ft of 6-in. pipe.

Determine:

Determine the acceleration head

Solution:

(1) Determine the average velocity in the 4-in. pipe

$$V_4 = \frac{(0.321)(80)}{(12.73)} = 2.02\,\text{fps}$$

(2) Determine the average velocity in the 6-in. pipe

$$V_6 = \frac{(0.321)(80)}{(28.89)} = 0.89\,\text{fps}$$

(3) Determine the acceleration head in the 4-in. pipe

$$H_{A4} = \frac{(4)(2.02)(360)(0.066)}{(1.4)(32.2)} = 4.26\text{ ft}$$

(4) Determine the acceleration head in the 6-in. pipe

$$H_{A6} = \frac{(20)(0.89)(360)(0.066)}{(1.4)(32.2)} = 9.38\,\text{ft}$$

(5) Total acceleration head

$$\begin{aligned} H_A &= H_{A4} + H_{A6} \\ &= 4.26 + 9.38 \\ &= 13.64\,\text{ft} \end{aligned}$$

2.2.2.17 Power requirements

The hydraulic horsepower (HHP) for a centrifugal pump is a theoretical value calculated from the rated capacity and differential head, assuming a 100% efficient pump. It can be calculated as:
Field units

$$HHP = \frac{(TDH)(\rho)Q}{550} \tag{2.25a}$$

SI units

$$HHP = \frac{(TDH)(\rho)Q}{367{,}000} \tag{2.25b}$$

where

HHP = hydraulic horsepower, hp.; 1 hp =550 ft lb/s) (kW)
TDH = pump head, ft (m)
ρ = density of liquid, lb/ft^3 (kg/m^3)
Q = flow rate, ft^3/s (m^3/h)

By making the appropriate unit conversions, Eqs. (2.25a), (2.25b) may be expressed as:
For kinetic energy pumps
Field units

$$HHP = \frac{(SG)(q)(TDH)}{3960} \tag{2.26a}$$

SI units

$$HHP = \frac{(SG)(q)(TDH)}{367.6} \tag{2.26b}$$

For positive displacement pumps
Field units

$$= \frac{(\text{gpm})(\Delta P)}{1714e} \tag{2.27a}$$

SI units

$$HHP = \frac{(m^3/h)(\Delta P)}{3600} \tag{2.27b}$$

Field units

$$HHP = \frac{(Q)(\Delta P)}{58{,}766} \tag{2.28}$$

where

HHP = hydraulic horsepower
ΔP = maximum working pressure of PD pumps, psig (kPa)

Q = flow rate, bpd (m^3/h)
q = flow rate, gpm (m^3/h)
SG = specific gravity relative to water

The input power to the shaft of the pump is called the brake horsepower and it is given by

$$BHP = HHP/e \tag{2.29}$$

where

BHP = brake horsepower
e = pump efficiency factor, obtained from the pump manufacturer or approximated from Fig. 2.21

For electric motor-driven pumps, the energy consumption can be estimated by

$$KWH/\text{day} = \frac{17.9(BHP)}{\text{Motor Efficiency}} \tag{2.30}$$

Note that changes in specific gravity will affect the power calculations but does not affect the head capability of a pump. The pump efficiency chart (Fig. 2.21) should be

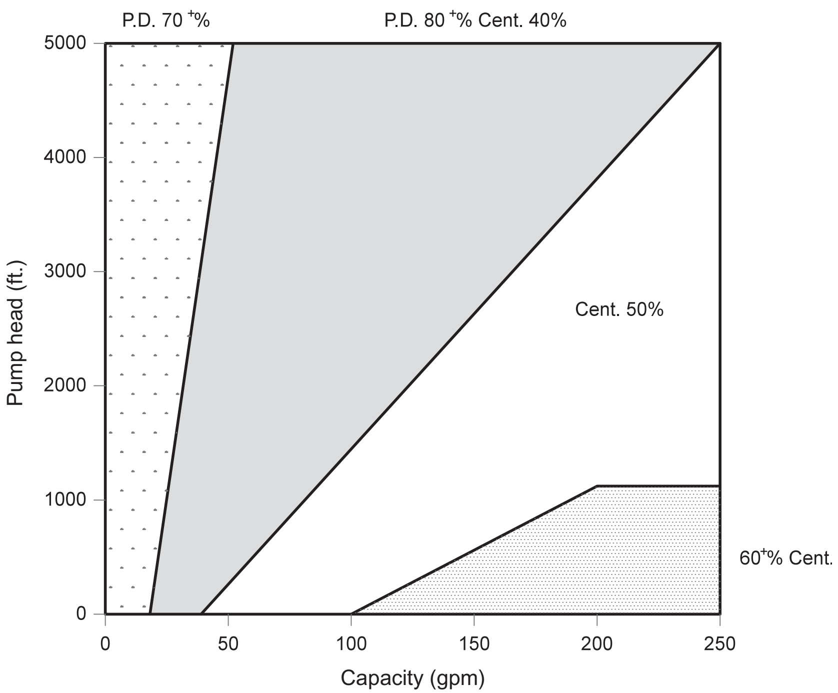


Fig. 2.21 Pump efficiency chart (for screening estimates only).

used for estimating purposes only. It is useful to determine whether a particular pumping application is best met by using a positive displacement or centrifugal pump. Positive displacement pumps offer high efficiencies at high TDHs and low flow rates. Centrifugal pumps offer better efficiencies at high flow rates and low TDH. For moderate TDHs and flows, either type could be used, depending on factors other than efficiency.

Example 2.2. Calculation of pump head and horsepower (field units)

Given:

(a) A pump takes suction from a vessel operating at 15 psig discharges to a tower operating at 500 psig.
(b) Fig. 2.22 and the following data:

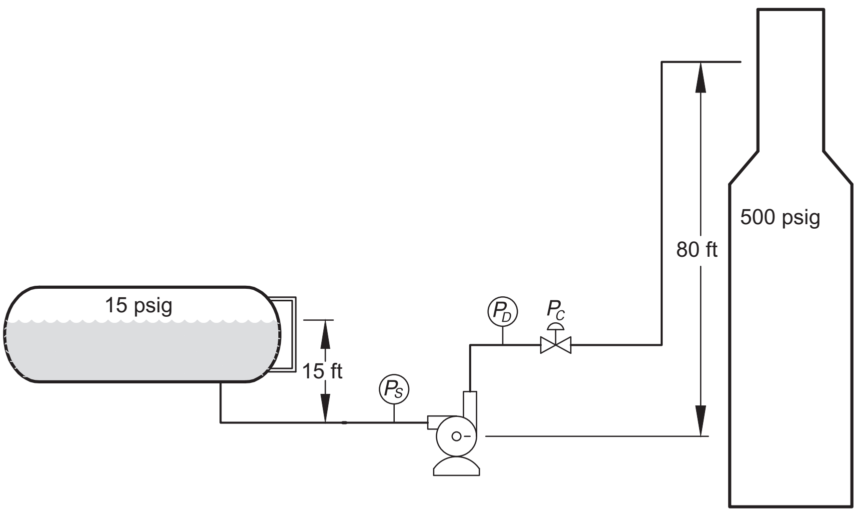


Fig. 2.22 Schematic for Example 2.2.

Flow	150 gpm
Specific gravity	0.8
Line losses	2 psi suction
Control losses	4 psi discharge
Static head	10 psi
	15 ft suction
	80 ft discharge
Atmospheric pressure	14.7 psia
Kinetic pump efficiency	0.52
PD pump efficiency	0.82
Motor efficiency	0.92

Determine:

Determine the following for both kinetic and positive displacement pumps:

(1) Pump suction head
(2) Pump discharge head
(3) Total dynamic head
(4) Pump horsepower required
(5) Electrical energy consumption

Solution:

1. Determine pump suction head

$$\begin{aligned} H_s &= \frac{(P_1 - P_{f1})(2.31)}{SG} + H_1 \\ &= \frac{(15-2)(2.31)}{0.8} + 15 \\ &= 52\,\text{ft}(\text{Kinetic}) \end{aligned}$$

or

$$= 52\left(\frac{0.8}{2.31}\right)$$

$$= 18\,\text{psig}(\text{Positive Displacement})$$

2. Determine pump discharge head

$$\begin{aligned} H_D &= \frac{(P_2 + P_{f2} + P_c)(2.31)}{SG} + H_2 \\ &= \frac{(500+4+10)(2.31)}{0.8} + 80 \\ &= 1564\,\text{ft}(\text{Kinetic}) \end{aligned}$$

or

$$= 1564\left(\frac{0.8}{2.31}\right)$$

$$= 542\,\text{psia}(\text{Positive Displacement})$$

3. Determine the total dynamic head

$$\begin{aligned} TDH &= H_D - H_s \\ &= 1564 - 52 \\ &= 1512\text{ft}(\text{Kinetic}) \end{aligned}$$

or

$$= 1512\left(\frac{0.8}{2.31}\right)$$

$$= 524\,\text{psig}(\text{Positive Displacement})$$

4. Determine the pump horsepower required

$$\begin{aligned} BHP &= \frac{(\text{gpm})(TDH)(SG)}{3960e} \\ &= \frac{(150)(1512)(0.8)}{3960(0.52)} \\ &= 88(\text{Kinetic}) \end{aligned}$$

or

$$\begin{aligned} BHP &= \frac{(\text{gpm})(\Delta P)}{1714e} \\ &= \frac{(150)(524)}{1714(0.82)} \\ &= 56(\text{Positive Displacement}) \end{aligned}$$

5. Determine the electrical energy consumption

$$\begin{aligned} KWHR/\text{Day} &= \frac{(17.9)(BHP)}{\text{Motor Efficiency}} \\ &= \frac{(17.8)(88)}{0.92} \\ &= 1.712(\text{Kinetic}) \end{aligned}$$

or

$$\begin{aligned} KWHR/\text{Day} &= \frac{(17.9)(56)}{0.92} \\ &= 1090(\text{Positve Displacement}) \end{aligned}$$

2.3 Selection criteria

2.3.1 General considerations

The first decision one must make when selecting a pump is to determine the type of pump to use. The "pump type" can be either centrifugal, reciprocating, rotary, metering, seal-less, or miscellaneous. This section discusses the advantages, features, and limitations of each pump type. Once a pump type is selected, one should refer to the respective chapter in this volume for a detailed discussion.

2.3.2 Selection fundamentals

Generally, centrifugal pumps should usually be considered first. The reason for this is due to their low cost, simplicity, reliability, and nonpulsating flow characteristics. However, under the following conditions, reciprocating or rotary pumps may be more applicable:

- *Very high head and low capacity*. A positive displacement (reciprocating or rotary) pump is designed for these conditions. However, centrifugal pumps of special design (high speed) have been built to pump as little as 10 to 20 gpm (against pressures of 1000 psi or more).
- *Low speed drivers*. A centrifugal pump for normal oil service operates at nominal speeds of 1800 or 3600 rpm. A pump driven by an internal combustion engine operating at 600 rpm will require a speed increaser. A gearbox can easily cost more than the pump. In this case, a positive displacement pump may prove more economical.
- *High efficiency requirements over a range of pressure conditions*. This situation frequently occurs in pipeline transportation. A positive displacement pump at any capacity remains reasonably efficient at pressures between 25% and 100% of its rating. A centrifugal pump will not remain efficient in these circumstances.

In pipeline service, centrifugal pumps usually are connected in series while reciprocating pumps are connected in parallel. These arrangements are most suitable to the pump's characteristics.

- *High viscosity*. The efficiency of centrifugal pumps starts to fall significantly when the viscosity of the oil exceeds 100 cSt or 500 SSU. While centrifugal pumps are used successfully at higher viscosities, especially where the pump is large, rotary or reciprocating pumps will probably be more economical.
- *Emulsification*. Impellers impose great agitation on pumped stock, making centrifugal pumps a poor choice where water/oil emulsions must be avoided. Both rotary and reciprocating pumps are better equipped to prevent emulsification.

When selecting a pump for a given service, conflicting factors may make the choice unclear. For example, if a pumped fluid is both highly viscous and contains abrasives, a rotary pump may be the first choice for the viscous fluid. However, because of the abrasives, ultimately it would be a poor choice.

2.3.2.1 Pump selection charts

Pump selection charts, Fig. 2.23, provide additional information that are helpful in choosing between centrifugal, reciprocating, and rotary pumps. The pump selection charts show the type of pump most suitable from a head-capacity standpoint. The pump selection charts can be used for initial selection of a pump type. Generally, select a centrifugal pump unless:

(a) Capacity at pumping head falls within area labeled "Positive Displacement Pumps."
(b) Viscosity is >1000 cp at pumping temperature.
(c) Percentage of undissolved gas by volume is $>5\%$ or gas is not well dispersed.
(d) The percentage of solids by volume is $>50\%$ or solids are not well dispersed.
(e) The pump is expected to run dry without automatic shutdown provisions, for example, pump out pumps.

2.3.2.2 Pump selection guide

The *pump selection guide*, Table 2.3, is used to identify other factors affecting the pump selection. The local priority column on the Pump Selection Guide weighs

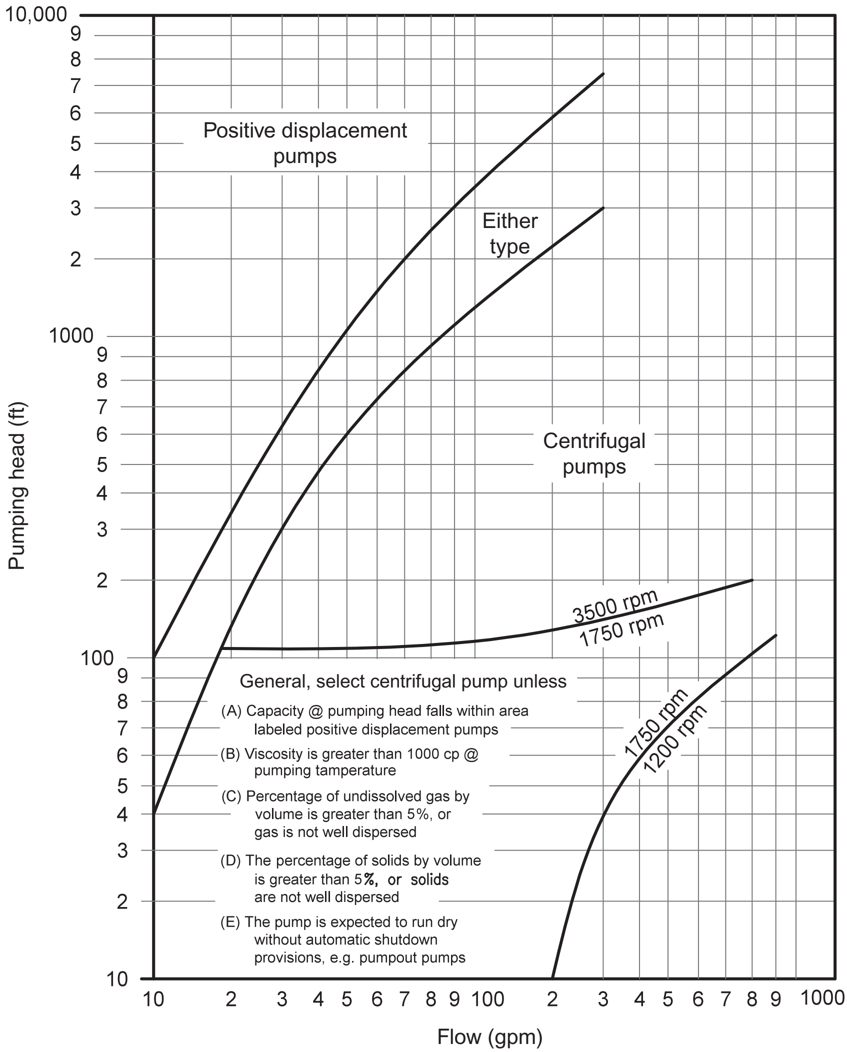


Fig. 2.23 Pump selection chart (1 of 2).
Courtesy of Ingersoll-Rand.

factors by their importance in a specific application. The designer should also consider operations preference, spare parts availability, and local maintenance capabilities.

2.3.3 Centrifugal pumps

Centrifugal pumps are used in approximately 80% of all production operations. There are many reasons why centrifugal pumps are used more often than any other pump type. Some of these reasons are as follows:

Table 2.3 Pump selection guide

Selection criteria	Local priority			Reciprocating			Centrifugal			Rotary			Comments
	Critical	Import	Not imp	Good	Fair	Poor	Good	Fair	Poor	Good	Fair	Poor	
Flow rate vs developed head				—See head/capacity selection curve—									
Avoids emulsifying the fluid				X					X	X			
Reliability						X	X				X		
Self-priming				X					X	X			Self-priming centrifugal designs are available
Handling abrasives					X			X				X	“Progressive cavity” rotary pumps handle abrasives
Handling low $NPSH_A$						X	X				X		$NPSH_A$ on Recip’s must include allowance for “acceleration head”
Handling entrained gas					X			X		X			
Pump viscous stock					X				X	X			
Pump low viscosity stock				X			X					X	
Energy efficiency				X				X			X		Centrifugal efficiency varies with type of pump, and flow rate relating to BEP
Installed cost					X		X			X			
Maintenance cost						X	X				X		

Table 2.3 Continued

Selection criteria	Local priority: Critical	Local priority: Import	Local priority: Not imp	Reciprocating: Good	Reciprocating: Fair	Reciprocating: Poor	Centrifugal: Good	Centrifugal: Fair	Centrifugal: Poor	Rotary: Good	Rotary: Fair	Rotary: Poor	Comments
Controls leakage to atmosphere						X	X (IF SEALED)			X (IF SEALED)			Air quality permits may dictate pump selection
Temperatures above 350°F													Varies with application. review each specific application
Client preferences													Many clients have strong preferences based on operating experience
Compatibility with existing system													Consider maintenance facilities, parts, and operator familiarity
Weight and space													Very important in offshore or other limited space applications. Review each specific application

- Lowest initial Capital Expense (CAPEX) and lowest Operating Expense (OPEX)
- Very reliable
- Capable of achieving a wide range of flow rates (10 to over 100,000 gpm)
- Capable of developing a wide range of heads (20 to over 10,000 ft)
- Ability to operate with relatively low $NPSH_A$
- Wide temperature and pressure capabilities
- Minimal environmental impact with proper seals
- Flow variation flexibility
- Easily controlled
- Available from many manufacturers
- Available in many different materials of construction
- Available in a wide range of designs and types
- Experiences low vibration
- Generates low pressure pulsations
- Capable of handling suspended solids

Limitations of centrifugal pumps include:

- *When a self-priming pump is needed*. Conventional centrifugal pumps must be primed and supplied with adequate NPSH for proper application. There are some centrifugal pumps available with a self-priming design. However, the available sizes are limited and performance penalties are substantial.
- *Low flow, moderate to high head applications*. Operating centrifugal pumps to the left of their best efficiency point (BEP) causes many problems. For example, impeller eye recirculation and high levels of vibration result when operating below the recommended minimum flow. The effects of low flow operation are shaft breakage, seal and bearing failures.

The definition of "minimum flow" is pump specific. Some of the factors that affect the minimum flow are fluid density, $NPSH_A$, impeller design suction specific speed, casing volute design, piping and control systems designs. Sundyne vertical in-line pumps are designed for low-flow, moderate to high head applications. However, these pumps will also have a minimum flow that should be maintained for reliable service.

- *Handling fluids with entrained gas*. Centrifugal pumps can lose suction due to an excess amount of gas accumulation in the impeller eye. The vapor or gas accumulation can be caused by excessive cavitation, recirculation, or entrained gas.

The centrifugal pump impeller is a good centrifuge. The heavy material (liquid) is expelled through the impeller while the light fluid (gas) will collect in the eye of the impeller. If the gas or vapor volume fills the impeller eye, the pump may lose suction.

A worst-case example of this would be a centrifugal pump selected for an application with little $NPSH_A$, some entrained gas, and operated at reduced flow rates. At reduced flows, the fluid velocity in the suction pipe may not be capable of pushing the entrained gas or vapor through the impeller.

- *Pumping viscous fluids*. Although centrifugal pumps are capable of pumping fluids with a viscosity of 880 cSt (4000 SSU) and higher, the performance penalties are substantial. As presented in Table 2.4, pump efficiency can be drastically reduced. The pump's capacity and head capability are also reduced.

Table 2.4 Centrifugal pump selection and application

		Typical capacity range (gpm)	Typical head range feet	Common applications
1	Vertical inline (single stage)	20 to 1200	15 to 600	General processing and transfer service at temperatures below 350°F. Minimum space application
2	End suction, frame mounted ANSI	35 to 4000	30 to 700	General processing and transfer service at temperatures below 350°F
3	Horizontal, single-stage, horizontal suction between bearings API	100 to 20,000	40 to 900	Hydrocarbons in the low to moderate flow and moderate head ranges. Cooling tower water circulation
4	Horizontal, multistage axially split	200 to 1500	200 to 4500	Used to pump crude oil, high-pressure boiler feed water, seawater, gasoline, and other hydrocarbons and also in waterflood operations
5	Horizontal barrel (double case)	200 to 1700	up to 9000	Used principally for process plant high-pressure reactor charge and waterflood applications. Pump speed may approach 7500 rpm
6	Radially split vertical can	20 to 2000	55 to 2000	Used principally for improving $NPSH_A$ when pumping bubble point hydrocarbon mixtures. Pumping end usually encased in a pressure vessel (can)
7	Vertical turbine lineshaft	100 to 30,000	10 to 1500	Used for lift application such as seawater and fire water
8	Electric submersible	100 to 30,000	10 to 2000	Used to eliminate long shaft lengths. Same application as vertical turbines and crude oil production

The selection of rotary or reciprocating pumps may be a better choice for low-flow, viscous services. However, if the required flow rate exceeds the capability of rotary or reciprocating pumps, centrifugal may be the only choice.

- *When fluid emulsification must be avoided.* Centrifugal pumps are good agitators and mixers. Rotary or reciprocating pumps are a better choice to minimize fluid emulsification.

2.3.4 Reciprocating pumps

Reciprocating pumps are used most often for low-capacity, high-pressure service. The initial cost of small reciprocating pumps is competitive with centrifugal pumps. However, larger reciprocating pumps (>200 gpm) are usually more expensive (initially and to maintain) than other pump types. High-speed centrifugal pumps should not be overlooked when low-capacity high head services are encountered. Table 2.5 compares typical positive displacement applications.

When service requirements allow using either a centrifugal or a reciprocating pump, one should carefully consider both operating and maintenance costs. For most services, the operating costs of motor- or turbine-driven centrifugal pumps are less than the costs for reciprocating pumps. Maintenance costs usually exceed those of a centrifugal pump because of the many moving parts, including valves and sliding contacts.

Pulsating flow may limit the use of reciprocating pumps. However, pulsating flow is usually not the decisive factor for determining if reciprocating pumps are the best selection. The effects of pulsation can be minimized but not eliminated by using pulsation dampers. Pulsating flow may cause problems in the application of automatic control flow measurement or process.

Typical applications for reciprocating pumps include:

- *Low to moderate capacity with high differential pressure*. Fig. 2.23 show the head-capacity range for which reciprocating pumps are normally considered in nonviscous services. The division shown is intended as a guideline only.
- *Relatively high viscosity*. The efficiency of a centrifugal pump drops rapidly with increasing viscosity. The most economical applications of these pumps are normally limited to viscosities under about 110 cSt (500 SSU). Reciprocating pumps can efficiently handle stocks up to about 1760 cSt (8000 SSU).

Higher viscosity oils can be delivered by reciprocating pumps operating at slower speeds, but such applications usually are not economical. Rotary pumps are more appropriate.

- Relatively constant capacity with widely varying discharge pressures. Reciprocating pumps are particularly suited to this application.
- Highly variable capacity with either constant or varying discharge pressure. Direct-acting, gas-driven pumps are well suited to this application because the speed is easily controlled by the gas driver.
- Where a self-priming pump is needed.

Limitations of reciprocating pumps include:

- *When pulsating flow is undesirable*. An example of such a service is a fuel oil feed to boilers. Ordinarily, rotary pumps are preferred for this service because of their smooth discharge pressure and better efficiency at higher viscosities.
- *Medium capacity and medium differential pressure with low viscosity*. Centrifugal pumps ordinarily are more economical. Examples of such services are water or hydrocarbons pumping about 100 gpm with differential pressures up to about 700 ft or 300 psi.

Table 2.5 Positive displacement pump selection and application

		Maximum flow (gpm)	Maximum pressure (psi)	Application
1	Plunger	300	6000	Used primarily as glycol pumps, steam generator feed pumps, condensate pumps, drilling mud pumps, and additive injection pumps, especially where solids are present
2	Piston	800	1500	Same as plunger, but without solids present
3	Diaphragm	10	1000	Used mainly in controlled volume applications as a metering pump or where low flow rates and high solids concentrations are present. Temperatures limited to <500°F maximum
4	Cam-and-piston			Vacuum services
5	Rotary gear	150	700	Used to transfer recovered oil from a drain separator to a process oil/water separator. Also used as a diesel transfer pump and in high-viscosity applications, and for circulating lubrication oil services for large machines and other clean, high- and low-pressure services
6	Single screw or progressive cavity	450	200	Used for high-viscosity stocks (non-Newtonian), stocks with up to 30% (entrained) gas, and often for abrasive services. Not good for temperatures over 300°F and in carbon dioxide entrained gas services
7	Three-screw	1000	4500	Used for low-temperature, high-viscosity stocks (lower then progressive cavity) without abrasives and circulating lubrication oil for large machines
8	Two or twin-screw	8000	4500	Used for low- and high-temperature liquids with and without abrasives (up to 600°F), and as a multiphase pump (90% plus gas, the rest liquid) with and without abrasives with special design
9	Sliding vane			Vacuum services

(1) For reciprocating pumps, pumps 1, 2, and 3 above. The recommended rpm is a function of stroke length (inches). See API Standard 674.
(2) For reciprocating pumps, pumps 1, 2, and 3 above, handling liquids with viscosities of 300 SSU at pumping temperature, the speeds are normally reduced to a percent of the basic speed. See API Standard 674.

- *High capacity*. For capacities above about 200 gpm, reciprocating pumps are seldom the best selection, regardless of the discharge pressure. In this range, reciprocating pumps become so large that they are more expensive than centrifugal pumps. An exception may be high-pressure water injection services.
- *Minimum packing leakage required*. Some hazardous or toxic services require absolute minimum stock leakage. Reciprocating pumps are subject to packing leaks and in such services must be fitted with a double stuffing box to provide an enclosed leakage disposal system. Centrifugal pumps with mechanical seals would be preferred.

2.3.5 Rotary pumps

- The most common rotary pumps are gear, multiple screw, and single screw. Cam-and-piston and sliding vane pumps can be considered for special services. The following discussion applies to all rotary pumps. Refer to Table 2.5 for positive displacement applications.

Typical applications for rotary pumps include

- *Viscous fluids*. Rotary pumps can deliver high-viscosity fluids with a smaller reduction in efficiency than other pump types. Rotary pumps can handle fluids with viscosities varying from LPG (not recommended for continuous service) to very viscous greases. For fluids with viscosities >2200 cSt (10,000 SSU), rotary pumps are usually the most economical selection.

Special reciprocating pumps operating at greatly reduced speeds can handle viscosities as high as those handled by rotary pumps, but these pumps must be so large, they become prohibitively expensive. Under specific conditions, centrifugal pumps can deliver viscosities up to about 1100 cSt (5000 SSU), but their efficiency above 110 cSt (500 SSU) is so poor that such applications are not economical.

- *Lubricating and hydraulic oils*. Rotary pumps are most commonly used to circulate lubricating oil in mechanical equipment or to provide pressure for hydraulic operating systems. The oil used in these systems is usually cleaned by filtering. The pumped fluid lubricates the pump's internal gears and bearings.
- *Self-priming*. Rotary pumps work well when services require self-priming in moderate capacities, such as barrel, small tank, and sump unloading. However, when these services involve high capacities and low-viscosity fluids, vertical centrifugal pumps are usually used.
- *Vacuum services*. Rotary pumps, lubricated by special oils, are often used in vacuum services to pump air or other gases and vapors. Low vapor pressure oils are used to lubricate the pumps and seal the clearance spaces. Oil separators on the discharge of such pumps remove the oil from the gas. Cam-and-piston type and sliding vane-type rotary pumps are commonly used in this manner. Pressures as low as 2×10^{-4} mm mercury absolute are attainable with the cam-and-piston type.

With either the oil or water seal, the vacuum obtainable is limited by the vapor pressure of the sealing liquid.

- *Intermittent low-capacity services*. Small internal-bearing rotary pumps can sometimes be used economically in intermittent services where rotary pumps might seem to be unsuited. In these cases, it is cheaper to periodically replace inexpensive pumps than it is to buy pumps not subject to the same rate of wear.

An example of this application is pumping out small tanks or vessels where a certain amount of scale and grit is expected. Another example is intermittent handling of non-lubricating fluids, such as LPG and gasoline on tank trucks.

- *Nonpulsating flow*. Hydraulic operating systems and fuel oil systems usually require non-pulsating flow. Rotary pumps work well in these services, especially when high viscosity renders centrifugal pumps uneconomical.
- *Handling "wet" oil*. "Wet" oils (>3% water by volume) should be used only with pumps that will not cause oil to emulsify. Rotary pumps operated at slow speed (300 to 400 rpm) work well in such services.

Limitations of rotary pumps include:

- *Nonlubricating fluids in continuous service*. Internal parts of rotary pumps must be adequately lubricated. Fluids with poor lubricating qualities, such as LPG, gasoline, and water are not usually satisfactory for rotary pumps in continuous service.
- High differential pressures and large capacity. Rotor deflection usually limits the differential pressure produced by a rotary pump. For standard designs, the larger the pump, the lower the maximum allowable differential pressure.
- *Medium-capacity and medium-head services*. Except for high viscosities, medium-capacity and medium-head services usually can be handled more economically by centrifugal or reciprocating pumps than by rotary pumps. Because of the rotary pump's close clearances and the possibility of mechanical damage, reciprocating and centrifugal pumps are usually recommended unless a rotary pump promises significant savings.
- *Abrasive material or possibility of running dry*. Rotary pumps are ordinarily not recommended for fluids containing appreciable quantities of abrasive material. However, under certain conditions, single-screw pumps with rubber liners (progressive cavity) can be used, although they may have very high maintenance requirements. In any case, rotary pumps should never be allowed to run dry.

Rotary pumps do not handle corrosive liquids well. Pumps constructed of brass and bronze are used occasionally. Stainless steel rotary pumps are not practical because of the possibility of galling or seizure.

2.3.6 Miscellaneous pumps

Table 2.6 lists pumps for special applications that do not fit into the typical centrifugal or positive displacement type.

2.4 Exercises

1. For classification purposes, all pumps can be classified as: (a)______________________
(b)__________________
2. Match the following terms:
___velocity head
___hydrostatics
___acceleration head
___static lift

Table 2.6 Miscellaneous pumps

Pump type	Characteristics
Axial flow	Also called propeller pump. Used where large capacity and low head are required. Generally with a vertical configuration for lifting wastewater, effluent, and so on
Disc friction	Also called regenerative turbine pumps. Similar to centrifugal pumps except liquid is pressured by recirculation in the impeller vanes. A low-capacity, moderate to high head pump that can handle large amounts of gas or vapor. Pump efficiency is greatly affected by internal clearances. These pumps are usually unsatisfactory where abrasives are present
Metering	Small reciprocating plunger or diaphragm pumps used for accurate pumping of chemicals and additives. Pumping rates are normally measured in gallons per hour and are adjustable from zero to full pump rate. Capable of high discharge pressures
Diaphragm	An air-operated, versatile, utility pump normally using compressed air as the driving fluid. Useful in handling hazardous or abrasive materials, and in explosive environments. Smaller units are occasionally used in metering service
Jet	Also called eductors or ejectors. Jet pumps have no moving parts and use the venturi action of high velocity fluids through a nozzle to create suction. The driving fluid and pumped fluid are mixed at the discharge. Typical applications are moving granulated solids with water, deep-well water pumping, and shipboard bilge pumping with water
Archimedes screw	Used in lifting effluent and waste water at relatively low flows where agitation and mixing are undesirable. Limited to lifts of approximately 25 ft. Similar to a screw conveyor
Peristaltic	Also called hosepumps. Peristaltic pumps are used for pumping fluids such as waste sludges, lime and cement mortar, adhesives, and shear-sensitive fluids such as latex paints. In the petrochemical industries use is limited to shear-sensitive services. The pumps have few moving parts, no seals, and can be run dry. Life is limited due to the life of the elastomer hose

___$NPSH_A$
___head
___NPSH margin
___static head
___$NPSH_R$
___hydrodynamics
___total dynamic head

- **(a)** the study of fluids at rest
- **(b)** the vertical distance from the surface of the fluid at rest to the pump datum line when the supply is below the datum line
- **(c)** $V^2/2g$
- **(d)** describes the losses associated with pulsating flow rates
- **(e)** difference between discharge head and suction head

(f) suction head minus vapor pressure
(g) the study of fluids in motion
(h) determined by test or calculation by pump manufacturer for a particular pump
(i) the vertical distance from the surface of a fluid at rest to the pump datum line, when the supply is above the datum line
(j) $NPSH_A$ less $NPSH_R$
(k) energy content of fluid per unit of weight referred to any arbitrary datum

3. Viscosity has a significant effect on pumps ________________
(a) warm-up
(b) efficiency
(c) head capability
(d) all the above
(e) A and B only

4. Describe the sound made by a cavitating pump.

____

5. When an existing pump exhibits insufficient NPSH margin, the _____________ should be checked.
(a) discharge backpressure valve
(b) lube oil level
(c) suction line
(d) low rate
(e) A, B, and D only

6. For high TDHs and low flow rates a ________ pump is preferred, and for high flow rates and low TDHs, a _________ pump is advisable.

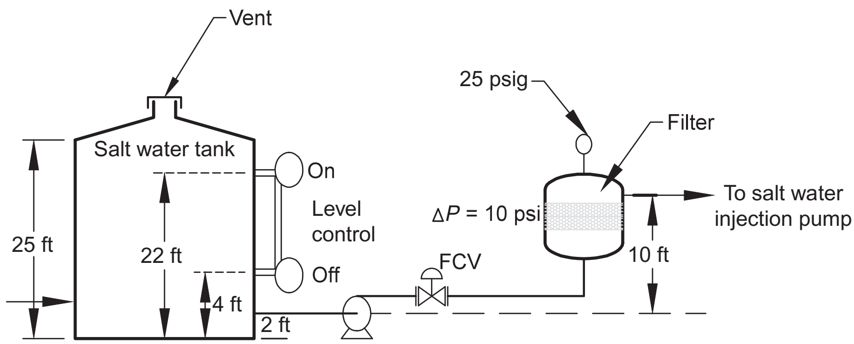
7. Head losses are due to:
(a) frictional flow resistance in the pipe
(b) valves
(c) fittings
(d) all the above
(e) none of the above

8. To increase $NPSH_A$
(a) use a booster pump
(b) raise the liquid level in the suction vessel
(c) raise the pump elevation
(d) use several suction sources
(e) A and B only

Questions 9–12

A centrifugal pump is used to pump salt water from an atmospheric storage tank with a required discharge pressure of 25 psig downstream of a filter. Use Fig. 2.24.

9. What is the minimum pump suction head? (Hint: at lowest tank level.)
(a) 0.5 ft
(b) 0.9 ft
(c) 6.3 ft
(d) 18.5 ft
(e) none of the above



Atmospheric pressure = 14 psia (=1300 ft elevation)
Fluid specific gravity = 1.05
Line losses = 0.5 psi suction
= 3 psi discharge
Flow = 87.5 gpm
Control losses = 5 psi discharge
Pump effeciency = 0.62

Fig. 2.24 Schematic for Problem 12.

10. What is the pump discharge head?
- **(a)** 3.1 ft
- **(b)** 23.7 ft
- **(c)** 82.6 ft
- **(d)** 104.6 ft
- **(e)** none of the above

11. What is the total dynamic head?
- **(a)** 16.7 ft
- **(b)** 42.6 ft
- **(c)** 59.9 ft
- **(d)** 103.7 ft
- **(e)** none of the above

12. What is the required pump horsepower?
- **(a)** 3.06 BHp
- **(b)** 3.88 BHp
- **(c)** 5.37 BHp
- **(d)** 8.24 BHp
- **(e)** 11.32 BHp

13. What are the two major groups that pumps fall into? Name an example from each group.

14. What are pumps commonly rated by?

15. True or false. Centrifugal pumps use pressure to build velocity following Bernoulli's 18th century hydraulic system laws.

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