

SECTION 12

Pumps & Hydraulic Turbines

Pumps

The most common types of pumps used in gas processing plants are centrifugal and positive displacement. Occasionally regenerative turbine pumps, axial-flow pumps, and ejectors are used.

Modern practice is to use centrifugal rather than positive displacement pumps where possible because they are usually less costly, require less maintenance, and less space. Conventional centrifugal pumps operate at speeds between 1200 and 8000 rpm. Very high speed centrifugal pumps, which can operate up

FIG. 12-1
Nomenclature

A = cross-sectional area of plunger, piston, or pipe, mm ²	S = suction specific speed (units per Equation 12-7)
a = cross-sectional area of piston rod, mm ²	sp gr = specific gravity at average flowing conditions. Equal to RD
AC = alternating current	T = torque, N · m (Newton meters)
bbl = barrel (42 U.S. gallons or 0.158987 m ³)	t _r = temperature rise, °C
bkW = brake kilowatt	u = impeller peripheral velocity, m/s
C = constant (Fig. 12-19)	VE = volumetric efficiency, fraction
C _p = specific heat at average temperature, J/(kg · °C)	VE _o = overall volumetric efficiency
D = displacement of reciprocating pump, m ³ /h	VE _p = volumetric efficiency due to density change
DC = direct current	VE _l = volumetric efficiency due to leakage
d = impeller diameter, mm	V _{pd} = pulsation dampener volume, mm ³
e = pump efficiency, fraction	v = liquid mean velocity at a system point, m/s
g = 9.80665 m/s ² (acceleration of gravity)	z = elevation of a point of the system above (+) or below (−) datum of the pump. For piping, the elevation is from the datum to the piping centerline; for vessels and tanks, the elevation is from the datum to the liquid level, m.
H = total equipment head, m of fluid	
h = head, m of fluid pumped	
hyd kW = hydraulic kilowatts	
k = factor related to fluid compressibility (Fig. 12-19)	
K = type of pump factor (Equation 12-18)	
kPa = kilopascal	
kPa (abs) = kilopascal, absolute	
kPa (ga) = kilopascal, gage	
L = length of suction pipe, m	
L _s = stroke length, mm	
m = number of plungers or pistons	
NPPP = net positive pipe pressure, kPa (abs) (NPPP = P _x − P _{vp} > 0)	
NPSH = net positive suction head of fluid pumped, m	
NPSHA = NPSH available, m	
NPSHR = NPSH required, m	
n = speed of rotation, revolutions/minute (rpm)	
n _s = specific speed (See Fig. 12-2 for units)	
N = Polytropic exponent of charge gas. (For nitrogen, N = 1.4)	
ΔP = differential pressure, kPa	
P = pressure, kPa (abs) or kPa (ga)	
P _{vp} = liquid vapor pressure at pumping temperature, kPa (abs)	
Q = rate of liquid flow, m ³ /h	
r = ratio of internal volume of fluid between valves, when the piston or plunger is at the end of the suction stroke, to the piston or plunger displacement.	
RD = relative density of pumped fluid at average flowing conditions to water density at standard conditions (15°C, 1 atm)	
s = slip or leakage factor for reciprocating and rotary pumps	
	Greek:
	ρ = density at average flowing conditions, kg/m ³
	ρ _i = inlet density, kg/m ³
	ρ _o = outlet density, kg/m ³
	Δ = allowable pressure fluctuations as a percentage of mean pressure
	Subscripts:
	a = acceleration
	ave = with P, average pressure in pulsating flow
	bep = best efficiency point, for maximum impeller diameter
	c = compression
	d = discharge of pump
	dv = discharge vessel
	D = displacement
	f = friction
	i = inlet of equipment
	l = leakage
	max = with P, maximum acceptable peak pressure in pulsating flow
	min = with P, minimum acceptable valley pressure in pulsating flow
	o = outlet of equipment
	ov = overall
	p = pressure
	pd = pulsation dampener

FIG. 12-1 (Cont'd)
Nomenclature

r = rise
s = static, suction of pump, specific, or stroke
sv = suction vessel
v = velocity
vp = vapor pressure

w = water
x = point x in the inlet subsystem
y = point y in the outlet subsystem
1 = impeller diameter or speed 1
2 = impeller diameter or speed 2

DEFINITIONS OF WORDS AND PHRASES USED IN PUMPS AND HYDRAULIC TURBINES

Alignment: The straight line relation between the pump shaft and the driver shaft.

Casing, axially split: A pump case split parallel to the pump shaft.

Casing, radially split: A pump case split transverse to the pump shaft.

Cavitation: A phenomenon that may occur along the flow path in a pump when the absolute pressure equals the liquid vapor pressure at flowing temperature. Bubbles then form which later implode when the pressure rises above the liquid vapor pressure.

Coupling: A device for connecting the pump shaft to the driver shaft consisting of the pump shaft hub and driver shaft hub, usually bolted together.

Coupling, spacer: A cylindrical piece installed between the pump shaft coupling hub and driver shaft coupling hub, to provide space for removal of the mechanical seal without moving the driver.

Cutwater: The point of minimum volute cross-sectional area, also called the volute tongue.

Datum elevation: The reference horizontal plane from which all elevations and heads are measured. The pumps standards normally specify the datum position relative to a pump part, e.g. the impeller shaft centerline for centrifugal horizontal pumps.

Diffuser: Pump design in which the impeller is surrounded by diffuser vanes where the gradually enlarging passages change the liquid velocity head into pressure head.

Displacement: The calculated volume displacement of a positive displacement pump with no slip losses.

Double acting: Reciprocating pump in which liquid is discharged during both the forward and return stroke of the piston.

Duplex: Pump with two plungers or pistons.

Efficiency, mechanical: The ratio of the pump hydraulic power output to pump power input.

Efficiency, volumetric: The ratio of a positive displacement pump suction or discharge capacity to pump displacement.

Head: The flowing liquid column height equivalent to the flowing liquid energy, of pressure, velocity or height above

the datum, whose sum is the total head. Also used to express changes of energy such as the friction losses, the equipment total head and the acceleration head.

Head, acceleration: The head equivalent to the pressure change due to changes in velocity in the piping system.

HPRT: Hydraulic power recovery turbine.

Impeller: The bladed member of the rotating assembly of a centrifugal pump which imparts the force to the liquid.

NPSHA: The total suction absolute head, at the suction nozzle, referred to the standard datum, minus the liquid vapor absolute pressure head, at flowing temperature available for a specific application. For reciprocating pumps it includes the acceleration head. NPSHA depends on the system characteristics, liquid properties and operating conditions.

NPSHR: The minimum total suction absolute head, at the suction nozzle, referred to the standard datum, minus the liquid vapor absolute pressure head, at flowing temperature, required to avoid cavitation. For positive displacement pumps it includes internal acceleration head and losses caused by suction valves and effect of springs. It does not include system acceleration head. NPSHR depends on the pump characteristics and speed, liquid properties and flow rate and is determined by vendor testing, usually with water.

Pelton wheel: A turbine runner which turns in reaction to the impulse imparted by a liquid stream striking a series of buckets mounted around a wheel.

Recirculation control: Controlling the quantity of flow through a pump by recirculating discharge liquid back to suction.

Rotor: The pump or power recovery turbine shaft with the impeller(s) mounted on it.

Rotor, Francis-type: A reverse running centrifugal pump impeller, used in a hydraulic power recovery turbine, to convert pressure energy into rotational energy.

Run-out: The point at the end of the head-capacity performance curve, indicating maximum flow quantity and usually maximum brake power.

Runner: The shaft mounted device in a power recovery turbine which converts liquid pressure energy into shaft power.

Shut-off: The point on the pump curve where flow is zero, usually the point of highest total dynamic head.

Simplex: Pump with one plunger or piston.

Single acting: Reciprocating pump in which liquid is discharged only during the forward stroke of the piston.

Slip: The quantity of fluid that leaks through the internal clearances of a positive displacement pump per unit of time. Sometimes expressed on a percentage basis.

Surging: A sudden, strong flow change often causing excessive vibration.

Suction, double: Liquid enters on both sides of the impeller.

Suction, single: Liquid enters one side of the impeller.

Throttling: Controlling the quantity of flow by reducing the cross-sectional flow area, usually by partially closing a valve.

Triplex: Pump with three plungers or pistons.

Vanes, guide: A series of angled plates (fixed or variable) set around the circumference of a turbine runner to control the fluid flow.

Volute, double: Spiral type pump case with two cutwaters 180° apart, dividing the flow into two equal streams.

Volute, single: Spiral type pump case with a single cutwater to direct the liquid flow.

Vortex breaker: A device used to avoid vortex formation in the suction vessel or tank which, if allowed, would cause vapor entrainment in the equipment inlet piping.

to 23 000 rpm and higher, are used for low-capacity, high-head applications. Most centrifugal pumps will operate with an approximately constant head over a wide range of capacity.

Positive displacement pumps are either reciprocating or rotary. Reciprocating pumps include piston, plunger, and diaphragm types. Rotary pumps are: single lobe, multiple lobe, rotary vane, progressing cavity, and gear types. Positive displacement pumps operate with approximately constant capacities over wide variations in head, hence they usually are installed for services which require high heads at moderate capacities. A special application of small reciprocating pumps in gas processing plants is for injection of fluids (e.g. methanol and corrosion inhibitors) into process streams, where their constant-capacity characteristics are desirable.

Axial-flow pumps are used for services requiring very high capacities at low heads. Regenerative-turbine pumps are used for services requiring small capacities at high heads. Ejectors are used to avoid the capital cost of installing a pump, when a suitable motive fluid (frequently steam) is available, and are usually low-efficiency devices. These kinds of pumps are used infrequently in the gas processing industry.

Fig. 12-1 provides a list of symbols and terms used in the text and also a glossary of terms used in the pump industry. Fig. 12-2 is a summary of some of the more useful pump equations. Fig. 12-3 provides guidance in selecting the kinds of pumps suitable for common services.

EQUIPMENT AND SYSTEM EQUATIONS

The energy conservation equation for pump or hydraulic turbine systems comes from Bernoulli's Theorem and relates the total head in two points of the system, the friction losses between these points and the equipment total head. Elevations are measured from the equipment datum.

The total head at any system point is:

$$h = z + h_p + h_v = z + \frac{1000 \cdot P}{\rho \cdot g} + \frac{v^2}{2 \cdot g} \quad \text{Eq 12-1}$$

The system friction head is the inlet system friction head plus the outlet system friction head:

$$h_f = h_{fx} + h_{fy} \quad \text{Eq 12-2}$$

The equipment total head is the outlet nozzle total head minus the inlet nozzle total head. H is positive for pumps and negative for HPRTs:

$$H = h_o - h_i = z_o - z_i + \frac{1000 (P_o - P_i)}{\rho \cdot g} + \frac{v_o^2 - v_i^2}{2 \cdot g} \quad \text{Eq 12-3}$$

When the elevation and size of inlet and outlet nozzles are the same, the equipment total head (H) equals the difference of pressure heads.

When using any suction-and-discharge-system points, the following general equation applies.

$$z_x + \frac{1000 \cdot P_x}{\rho \cdot g} + \frac{v_x^2}{2 \cdot g} - h_{fx} + H = z_y + \frac{1000 \cdot P_y}{\rho \cdot g} + \frac{v_y^2}{2 \cdot g} + h_{fy} \quad \text{Eq 12-4}$$

When the points are located in tanks, vessels or low velocity points in the piping, the velocity head is normally negligible, but may not be negligible in equipment nozzles. Note that the subscripts "i" and "o" are used for variables at pumps and HPRTs inlet and outlet nozzles, respectively, while the subscripts "s" and "d" are used only for variables at pumps suction and discharge nozzles. The subscripts "x" and "y" are used for variables at points in each inlet and outlet subsystem and usually are suction and discharge vessels. Also "x" and "y" are used for friction head from point "x" to equipment inlet nozzle and from equipment outlet nozzle to point "y".

The work done in compressing the liquid is negligible for practically incompressible liquids and it is not included in the above equations. To evaluate the total head more accurately when handling a compressible liquid, the compression work should be included. If a linear relationship between density and pressure is assumed, the liquid compression head that substitutes for the difference of pressure heads in above equations is:

$$H_c = \frac{500 \cdot (P_o - P_i)}{g} \left[\frac{1}{\rho_o} + \frac{1}{\rho_i} \right] \quad \text{Eq 12-5}$$

When the differential pressure is sufficiently high to have a density change of more than 10%, or when the pressure is near the fluid's critical pressure, the change in fluid density and other properties with pressure is not linear. In these cases Equations 12-3 to 12-5 may not be accurate. A specific fluid properties relationship model is required in this case. For pure substances, a pressure-enthalpy-entropy chart may be used for estimating purposes by assuming an isentropic process. The pump manufacturer should be consulted for the real process, including the equipment efficiency, heat transfer, etc. to determine the equipment performance.

NET POSITIVE SUCTION HEAD

See NPSH definition in Fig. 12-1. There should be sufficient net positive suction head available (NPSHA) for the pump to work properly, without cavitation, throughout its expected capacity range. Usually a safety margin of about 0.6 to 1 m of

FIG. 12-2
Common Pump Equations

FLOW RATE						
Given ⇒ multiply by to get ↓	US gal/min	UK gal/min	ft ³ /sec	bbl/day	liters/s	Kg/h
m ³ /h	0.227 125	0.272 766	101.941	6.624 47 · 10 ⁻³	3.600 00	1/(999.102 · RD)
PRESSURE						
Given ⇒ multiply by to get ↓	lb/in ²	ft water at 39.2°F	m water at 0°C	m liquid	bar	* std atm 760 mm Hg at 0°C
kPa	6.894 76	2.988 99	9.805 11	ρ · g/1000	100	101.325
DENSITY						
Given ⇒ multiply by to get ↓	lb/ft ³	lb/US gal	lb/UK gal	kg/lt	API gravity	Baumé gravity
kg/m ³	16.0185	119.826	99.7763	1000	See Fig. 1-3	

$$h_p = \frac{P}{g \cdot RD \cdot 0.999\ 102} = \frac{1000 \cdot P}{g \cdot \rho} \quad \text{hyd kW} = \frac{Q \cdot H \cdot RD}{367.428} = \frac{Q \cdot H \cdot g \cdot \rho}{3\ 600\ 000} \quad \text{bkW} = \frac{Q \cdot H \cdot RD}{367.428 \cdot e} = \frac{Q \cdot H \cdot g \cdot \rho}{3\ 600\ 000 \cdot e}$$

$$h_v = \frac{v^2}{2g} = \frac{Q \cdot \Delta P^{**}}{3600} = \frac{Q \cdot \Delta P^{**}}{3600 \cdot e} \quad (\text{for pumps})$$

$$u = \frac{d \cdot n \cdot \pi}{60\ 000} \quad \text{bkW} = \frac{\text{hyd kW}}{e} \quad (\text{for pumps}) \quad T = \frac{(9549.30) (\text{bkW})}{n}$$

$$v = \frac{(Q) (277.778)}{A} \quad \text{bkW} = \text{hyd kW} \cdot e \quad (\text{for turbines})$$

$$1\ \text{HP} = 0.745\ 700\ \text{kW} \quad \text{RD} = \text{relative density}$$

$$= 550\ \text{ft} \cdot \text{lb}/\text{s} \quad \text{Water density at } 15^\circ\text{C} = 999.102\ \text{kg}/\text{m}^3$$

$$= 33,000\ \text{ft} \cdot \text{lb}/\text{min} \quad \text{Standard gravity acceleration:}$$

$$g = 9.806\ 65\ \text{m}/\text{s}^2 = 32.1740\ \text{ft}/\text{s}^2$$

$$n_s = \frac{n \cdot \sqrt{Q_{\text{bep}}}}{H_{\text{bep}}^{3/4}} = \frac{n \cdot H_{\text{bep}}^{1/4} \cdot \sqrt{Q_{\text{bep}}}}{H_{\text{bep}}}$$

See Fig. 1-7 for viscosity relationships
 * Standard atmospheric pressure:
 1 atm = 760 mm Hg = 101.325 kPa = 14.6959 psi
 ** See Equation 12-3 and 12-4.

CENTRIFUGAL PUMPS AFFINITY LAWS

1: Values at initial conditions
2: Values at new conditions

CHANGE ⇒	SPEED	DIAMETER	SPEED AND DIAMETER
$Q_2 =$	$Q_1 (n_2/n_1)$	$Q_1 (d_2/d_1)$	$Q_1 (d_2/d_1) (n_2/n_1)$
$h_2 =$	$h_1 (n_2/n_1)^2$	$h_1 (d_2/d_1)^2$	$h_1 [(d_2/d_1) (n_2/n_1)]^2$
$\text{bkW}_2 =$	$\text{bkW}_1 (n_2/n_1)^3$	$\text{bkW}_1 (d_2/d_1)^3$	$\text{bkW}_1 [(d_2/d_1) (n_2/n_1)]^3$
$\text{NPSHR}_2 =$	$\text{NPSHR}_1 (n_2/n_1)^2$	—	$\text{NPSHR}_1 (n_2/n_1)^2$

NPSHA above NPSHR is adequate. Cavitation causes noise, impeller damage, and impaired pump performance. Consideration must also be given to any dissolved gases which may affect vapor pressure. For a given pump, NPSHR increases with increasing flow rate. If the pump suction nozzle pressure is known

$$\text{NPSHA} = \frac{1000 \cdot (P_i - P_{vp})}{\rho \cdot g} + z_i + \frac{v_i^2}{2 \cdot g} \quad \text{Eq 12-6a}$$

When the pump suction nozzle pressure is not known, but the pressure at any point (x) of the suction system is known, NPSHA may be calculated with the following equation, where h_{fx} is the head friction loss from the point x to the suction nozzle. With commonly used suction pipe diameters, the velocity head may be negligible.

$$\text{NPSHA} = \frac{1000 \cdot (P_x - P_{vp})}{\rho \cdot g} + z_x + \frac{v_x^2}{2 \cdot g} - h_{fx} \quad \text{Eq 12-6b}$$

Moreover, when the suction system point is the specific case of the suction vessel, the equation is the following, where h_{fv} is the head friction loss from the suction vessel to the suction nozzle.

$$\text{NPSHA} = \frac{1000 \cdot (P_{sv} - P_{vp})}{\rho \cdot g} + z_{sv} - h_{fsv} \quad \text{Eq 12-6c}$$

The pressures in the above equations must be both absolute or gage; when using gage pressure both must be relative to the same atmospheric pressure. To convert a system pressure gage reading to absolute pressure add the existing local atmospheric pressure. The fluid vapor pressure must be at operating temperature. If the fluid vapor pressure is given in gage pressure, check which atmospheric pressure is reported. The use of the true local atmospheric pressure is very important in the cases of high altitude locations, and of a close margin of NPSHA over the NPSHR.

The pressure shall be measured at the pipe or nozzle centerline height; otherwise, adequate correction shall be made. Pay special attention to large pipe or nozzle diameters and the elevation of gage attached to them, pole or panel mounted instrument elevation, and different density fluid in the instrument line, see Hydraulic Institute Standards.⁵

To avoid vapor formation in the suction system, there must also be a Net Positive Pipe Pressure (NPPP) along it. Therefore, for every suction line point and operating condition the line pressure, at the top of the pipe must be higher than the fluid vapor pressure, being the pressure determined taking into account the pipe elevation.

The entrained and dissolved air or gases in the pump suction affects the pump performance, both mechanically and hydraulically, especially when the suction nozzle pressure is lower than the suction vessel pressure. In centrifugal pumps it causes the reduction of capacity and discharge pressure, because of the reduced overall density; and also, at low flow, the impeller centrifugal action separates the gas from the liquid resulting in the cessation of the liquid flow. For these cases, specially designed centrifugal pumps with higher tolerance to gas entrainment or

rotary pumps may be considered, even though their capacity is affected by entrained and dissolved air or gases. See Hydraulic Institute Standards.⁵

Datum

The pump datum elevation is a very important factor to consider and should be verified with the manufacturer. Some common references are shown in Fig. 12-4. Some manufacturers provide two NPSHR curves for vertical can pumps, one for the first stage impeller suction eye and the other for the suction nozzle.

NPSH Correction Factors

NPSHR is determined from tests by the pump manufacturer using water near room temperature and is expressed in height of water. When hydrocarbons or high-temperature water are pumped, less NPSH is required than when cold water is pumped. Hydraulic Institute correction factors for various liquids are reproduced in Fig. 12-5. Some users prefer not to use correction factors to assure a greater design margin of safety.

NPSH and Suction Specific Speed

Suction specific speed is an index describing the suction capabilities of a first stage impeller and can be calculated using Equation 12-7. Use half of the flow for double suction impellers.

$$S = \frac{n\sqrt{Q_{bep}}}{\text{NPSHR}_{bep}^{3/4}} \quad \text{Eq 12-7}$$

Pumps with high suction-specific speed tend to be susceptible to vibration (which may cause seal and bearing problems) when they are operated at other than design flow rates. As a result, some users restrict suction specific speed, and a widely accepted maximum is 11,000. For more details on the significance

FIG. 12-3
Pump Selection Guide

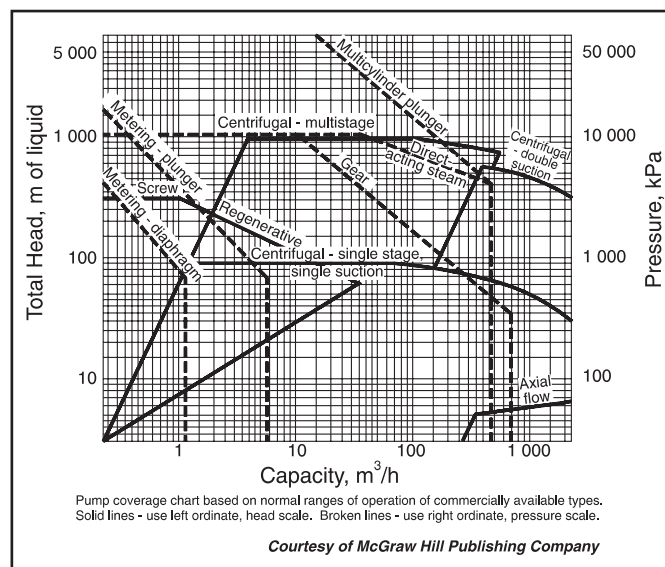
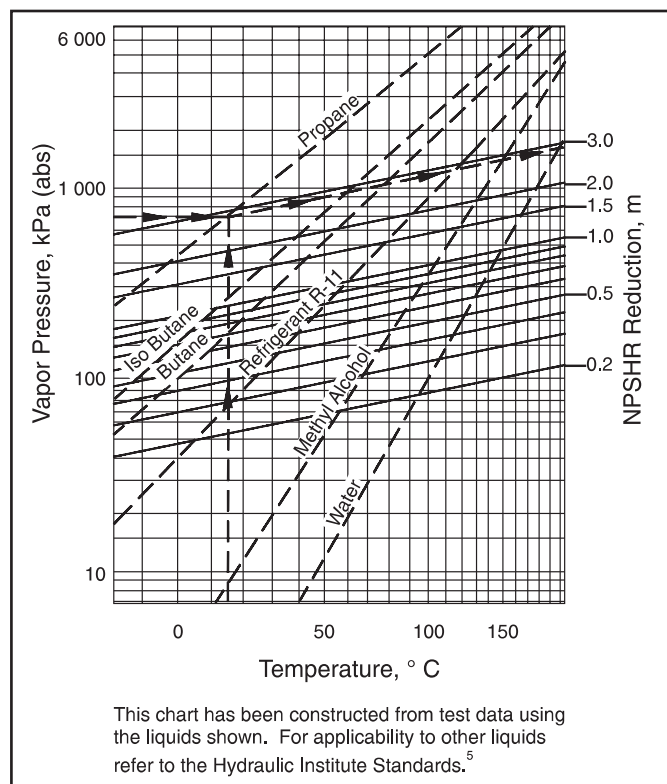


FIG. 12-4
Datum Elevation

Pump type	Standard	Datum elevation
Centrifugal, horizontal	API 610 ¹ Hydraulic Institute ⁵	Shaft centerline
Centrifugal, vertical in-line	API 610 ¹	Suction nozzle centerline
Centrifugal, other vertical	API 610 ¹	Top of the foundation
Centrifugal, vertical single suction, volute and diffused vane type	Hydraulic Institute ⁵	Entrance eye to the first stage impeller
Centrifugal, vertical double suction	Hydraulic Institute ⁵	Impeller discharge horizontal centerline
Vertical turbine. Line shaft and submersible types	AWWA E101 ¹⁸	Underside of the discharge head or head baseplate
Reciprocating	Hydraulic Institute ⁵	Suction nozzle centerline
Rotary	Hydraulic Institute ⁵	Reference line or suction nozzle centerline

FIG. 12-5

NPSHR Reduction for Centrifugal Pumps Handling Hydrocarbon Liquids and High Temperature Water



of suction specific speed, consult pump vendors or references listed in the References section.

Submergence

The suction system inlet or the pump suction bell should have sufficient height of liquid to avoid vortex formation, which may entrain air or vapor into the system and cause loss of capacity and efficiency as well as other problems such as vibration, noise, and air or vapor pockets. Inadequate reservoir geometry can also cause vortex formation, primarily in vertical submerged pumps. Refer to the Hydraulic Institute Standards⁵ for more information.

CALCULATING THE REQUIRED DIFFERENTIAL HEAD

The following procedure is recommended to calculate the head of most pump services encountered in the gas processing industry. See Example 12-1.

1. Prepare a sketch of the system in which the pump is to be installed, including the upstream and downstream vessels (or some other point at which the pressure will not be affected by the operation of the pump). Include all components which might create frictional head loss (both suction and discharge) such as valves, orifices, filters, and heat exchangers.

2. Show on the sketch:

- The datum position (zero elevation line) according to the proper standard. See Fig. 12-4.

- The pump nozzles sizes and elevations.
- The minimum elevation (referred to the datum) of liquid expected in the suction vessel.
- The maximum elevation (referred to the datum) to which the liquid is to be pumped.
- The head loss expected to result from each component which creates a frictional pressure drop at design capacity.

3. Use appropriate equations (Equations 12-1–12-4).
4. Convert all the pressures, frictional head losses, and static heads to consistent units (usually kPa or meters of head). In 5 and 6 below, any elevation head is negative if the liquid level is below the datum. Also, the vessel pressures are the pressures acting on the liquid surfaces. This is very important for tall towers. In the case of partitioned vessels, be sure to use the corresponding chamber pressure and liquid level elevation. And when the liquid is not a continuous phase, or it is not clear where the liquid level is, as in the case of packed fractionating towers, consider only the piping and exclude such vessels from the system.
5. Add the static head to the suction vessel pressure, then subtract the frictional head losses in the suction piping. This gives the total pressure (or head) of liquid at the pump suction flange.
6. Add the discharge vessel pressure, the frictional head losses in the discharge piping system, and the discharge static head. This gives the total pressure (or head) of liquid at the pump discharge. According to the type of capacity and head controls, pump type and energy conservation, required for the particular situation, provide a head and/or a flow additional margin to provide a good control. A control valve to throttle the discharge or to recirculate the flow, or a variable speed motor, etc. may be the options to provide good control.
7. Calculate the required pump total head by subtracting the calculated pump suction total pressure from the calculated pump discharge total pressure and converting to head.
8. It is prudent to add a safety factor to the calculated pump head to allow for inaccuracies in the estimates of heads and pressure losses, and pump design. Frequently a safety factor of 10% is used, but the size of the factor used for each pump should be chosen with consideration of:
 - The accuracy of the data used to calculate the required head
 - The cost of the safety factor
 - The problems which might be caused by installing a pump with inadequate head.

Example 12-1 — Liquid propane, at its bubble point, is to be pumped from a reflux drum to a depropanizer. The maximum flow rate is expected to be 82 m³/h. The pressures in the vessels are 1380 and 1520 kPa (abs) respectively. The relative density of propane at the pumping temperature (38°C) is 0.485. The elevations and estimated frictional pressure losses are shown on Fig. 12-6. The pump curves are shown in Fig. 12-7. The pump nozzles elevations are zero and the velocity head at nozzles is negligible.

