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

FIG. 12-1

Nomenclature

A = cross-sectional area of plunger, piston, or pipe, mm^2
a = cross-sectional area of piston rod, mm^2
AC = alternating current
bbl = barrel (42 U.S. gallons or 0.1589 m^3)
bkW = brake kilowatt
C = constant (Fig. 12-16)
C_p = specific heat at average temperature, J/(kg • °C)
D = displacement of reciprocating pump, m^3/h
DC = direct current
d = impeller diameter, mm
e = pump efficiency, fraction
g = 9.8067 m/s^2 (acceleration of gravity)
H = total equipment head, m of fluid
h = head, m of fluid pumped
hyd kW = hydraulic kilowatts
k = factor related to fluid compressibility (Fig. 12-16)
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
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, rpm
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 to water at standard temperature
s = slip or leakage factor for reciprocating and rotary pumps
S = suction specific speed (units per Eq 12-7)
sp gr = specific gravity at average flowing conditions. Equal to RD
T = torque, N • m (Newton meters)
t_r = temperature rise, °C
u = impeller peripheral velocity, m/s
VE = volumetric efficiency, fraction
VE_o = overall volumetric efficiency
VE_ρ = volumetric efficiency due to density change
VE_l = volumetric efficiency due to leakage
v = liquid mean velocity at a system point, m/s
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.

Greek:

ρ = density at average flowing conditions, kg/m^3
ρ_i = inlet density, kg/m^3
ρ_o = outlet density, kg/m^3

Subscripts:

a = acceleration
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
o = outlet of equipment
ov = overall
p = pressure
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 system
y = point y in the system
1 = impeller diameter or speed 1
2 = impeller diameter or speed 2

FIG. 12-1 (Cont'd)

Nomenclature

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 the 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 horsepower.

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.

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	0.2271	0.2728	101.9	$6.624 \bullet 10^{-3}$	3.6	1/(999 • 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	kgf/cm^2
kPa	6.895	2.989	9.8066	ρ • g/1000	100	101.325	98.066

DENSITY						
Given ⇒ multiply by to get ⇓	lb/ft^3	lb/US gal	lb/UK gal	kg/lt	API gravity	Baumé gravity
kg/m^3	16.018	119.83	99.77	1000	See Fig. 1-3	

$$h_p = \frac{P}{g \bullet RD \bullet 0.999} = \frac{1000 \bullet P}{g \bullet \rho}$$

$$h_v = \frac{v^2}{2\,g}$$

$$u = \frac{d \bullet n \bullet \pi}{60\,000}$$

$$v = \frac{Q \bullet 277.8}{A}$$

$$\text{hyd kW} = \frac{Q \bullet H \bullet RD}{367.47} = \frac{Q \bullet H \bullet g \bullet \rho}{3\,600\,000} = \frac{Q \bullet \Delta P}{3600}^{**}$$

$$\text{bkW} = \frac{\text{hyd kW}}{e} \quad \text{(for pumps)}$$

$$\text{bkW} = \text{hyd kW} \bullet e \quad \text{(for turbines)}$$

RD = relative density

Water density at 15°C = 999 kg/m^3

Standard gravity acceleration:

$g = 9.80665\ m/s^2 = 32.174\ ft/s^2$

$$\text{bkW} = \frac{Q \bullet H \bullet RD}{367.47 \bullet e} = \frac{Q \bullet H \bullet g \bullet \rho}{3\,600\,000 \bullet e} = \frac{Q \bullet \Delta P}{3600 \bullet e}^{**} \quad \text{(for pumps)}$$

$$T = \frac{9549 \bullet \text{bkW}}{n}$$

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

1 HP = 0.7457 kW
= 550 ft • lbf/s
= 33,000 ft • lbf/min

See Fig. 1-7 for viscosity relationships

*Standard atmospheric pressure:

1 atm = 760 mm Hg = 101.325 kPa = 14.696 psi

**See Eq. 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 =	$h_1\ (n_2/n_1)^2$	$h_1\ (d_2/d_1)^2$	$h_1\ [(d_2/d_1)\ (n_2/n_1)]^2$
bkW_2 =	$bkW_1\ (n_2/n_1)^3$	$bkW_1\ (d_2/d_1)^3$	$bkW_1\ [(d_2/d_1)\ (n_2/n_1)]^3$
$NPSHR_2$ =	$NPSHR_1\ (n_2/n_1)^2$	—	$NPSHR_1\ (n_2/n_1)^2$

up 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:

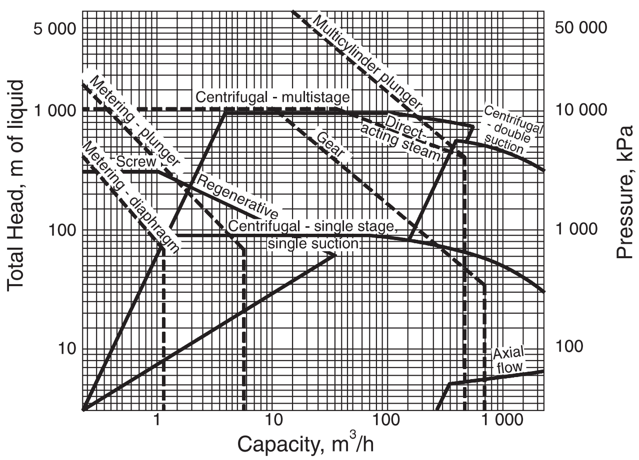
$$h = z + h_p + h_v = z + \frac{1000 \bullet P}{\rho \bullet g} + \frac{v^2}{2 \bullet g} \qquad \textbf{Eq 12-1}$$

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

$$h_f = h_{fx} + h_{fy} \qquad \textbf{Eq 12-2}$$

FIG. 12-3

Pump Selection Guide



Pump coverage chart based on normal ranges of operation of commercially available types.
Solid lines - use left ordinate, head scale. Broken lines - use right ordinate, pressure scale.

Courtesy of McGraw Hill Publishing Company

The equipment total head is the outlet nozzle total head minus the inlet nozzle total head:

$$H = h_o - h_i = z_o - z_i + \frac{1000\,(P_o - P_i)}{\rho \bullet g} + \frac{v_o^2 - v_i^2}{2 \bullet g} \qquad \textbf{Eq 12-3}$$

When the elevation and size of inlet and outlet nozzles are the same, the equipment total head (H) equals the difference of pressure heads. H is positive for pumps and negative for HPRTs.

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

$$z_x + \frac{1000 \bullet P_x}{\rho \bullet g} + \frac{v_x^2}{2 \bullet g} - h_{fx} + H = z_y + \frac{1000 \bullet P_y}{\rho \bullet g} + \frac{v_y^2}{2 \bullet g} + h_{fy}$$

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 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:

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

$$H_c = \frac{(P_o - P_i)}{2}\left[\frac{1}{\rho_o} + \frac{1}{\rho_i}\right] \qquad \textbf{Eq. 12-5}$$

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

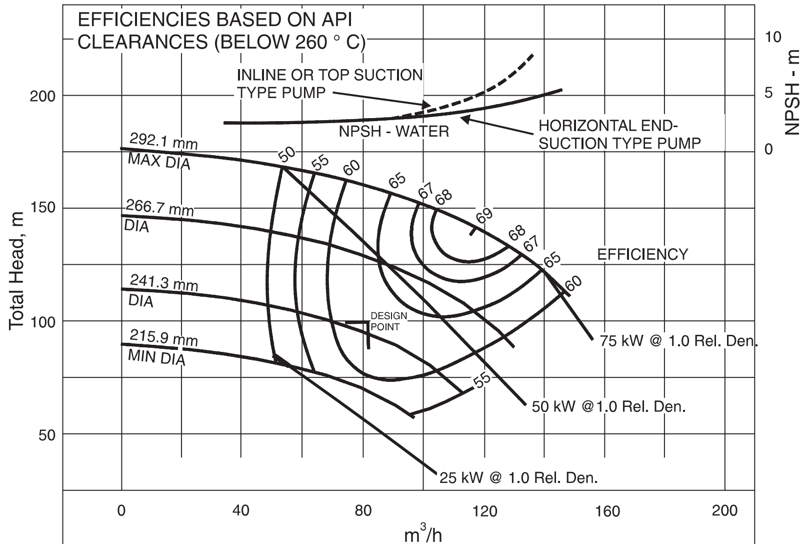
See NPSH definition in Fig. 12-1. There should be sufficient net positive suction head available (NPSHA) for the pump to work properly, without cavitation, throughout its expected capacity range. Usually a safety margin of about 0.6 to 1 m of NPSHA above NPSHR is adequate. Cavitation causes noise, impeller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

$$NPSHA = \frac{1000 \bullet (P_i - P_{vp})}{\rho \bullet g} + z_i + \frac{v_i^2}{2 \bullet g}$$

$$= \frac{1000 \bullet (P_{sv} - P_{vp})}{\rho \bullet g} + z_{sv} - h_{fsv} \qquad \textbf{Eq 12-6}$$

FIG. 12-5

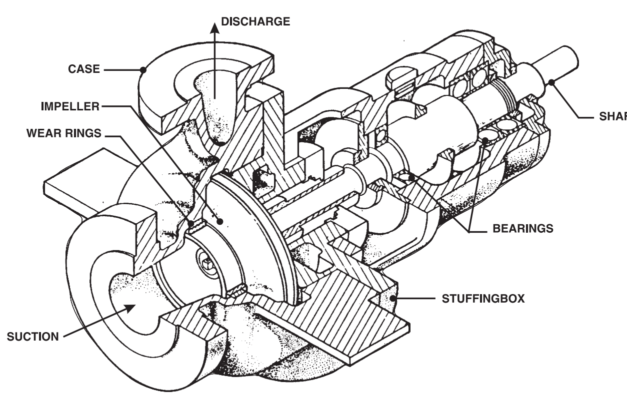
Depropanizer Reflux Pump for Example 12-1



DEPROPANIZER REFLUX PUMP					
IMPELLER MAX DIA	292.1	mm	3 X 4 X 11.5 SINGLE STAGE PUMP		
MIN DIA	215.9	mm	DIA IMPELLER 247.7 mm	IMPELLER PATT	3560 RPM
EYE AREA	91.6	SQ CM	NPSH REQUIRED 2.7 m	REFERENCE	CURVE NO. CA-307-3

FIG. 12-6a

Horizontal Single Stage Process Pump



Pump Type: Overhung Impeller, Separately Coupled, Single Stage, Centerline Support, API 610

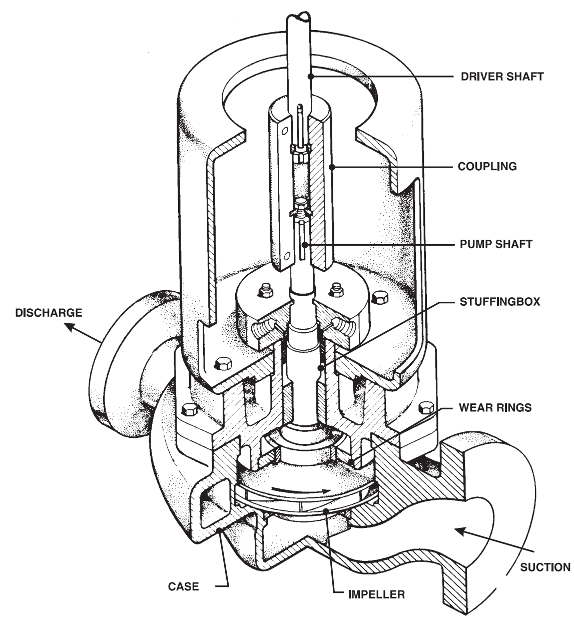
Range: 2 - 2300 m^3/h; 15 - 200 m TH; 1200 - 3600 rpm

Typical Application: Reflux, circulation, booster, boiler feed

Courtesy of Bingham – Willamette Ltd.

FIG. 12-6b

Vertical Inline Pump



Pump Type: Inline, Separately Coupled, Single Stage, Rigid Coupling, API 610

Range: 2 - 800 m^3/h; 15 - 200 m TH; 1200 - 3600 rpm

Typical Application: Reflux, circulation, booster, boiler feed

Courtesy of Bingham – Willamette Ltd.

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. 12-9. 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 Eq. 12-7. Use half of the flow for double suction impellers.

$$S = \frac{n\sqrt{Q_{bep}}}{NPSHR_{bep}^{3/4}} \qquad \textbf{Eq. 12-7}$$

Pumps with high suction 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 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 Standards[5] for more information.

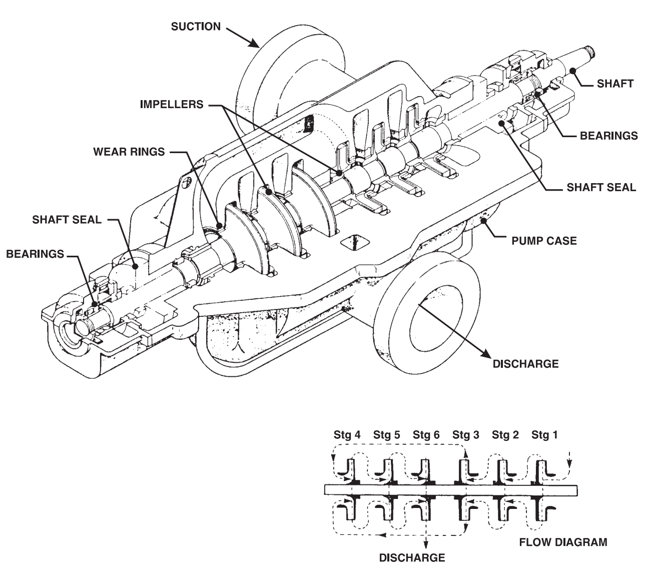
CALCULATING THE REQUIRED DIFFERENTIAL HEAD

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.

FIG. 12-6c

Horizontal Multi-Stage Pump



Pump Type: Horizontal, multi-stage, opposed impellers, axial split case, separately coupled

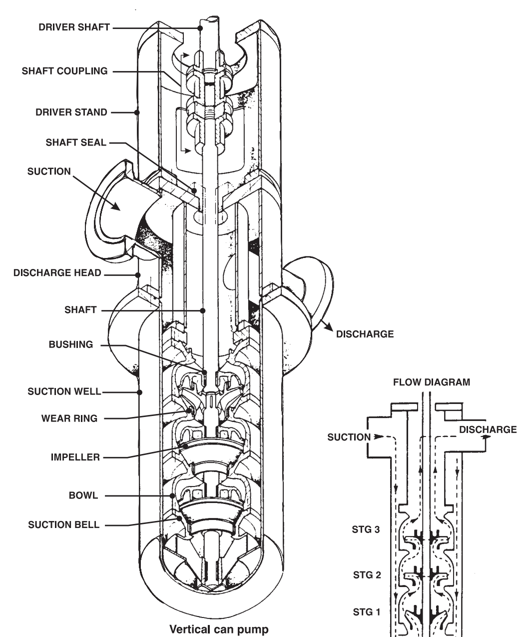
Range: 2 - 680 m³/h; 240 - 3050 m TH; 1800 - 7000 rpm

Typical Application: Treating solution charge, absorption oil, boiler feed, pipeline, water and NGL injection

Courtesy of Bingham – Willamette Ltd.

FIG. 12-6d

Vertical Can Pump



Pump Type: Vertical can, multi-stage, turbine

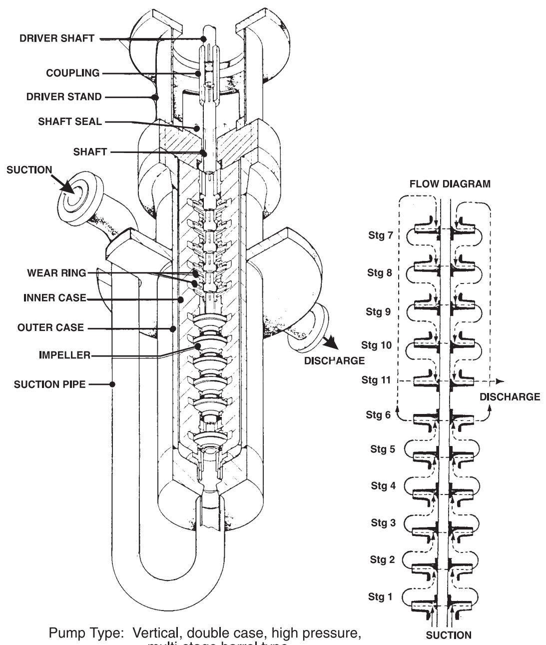
Range: 11 - 2300 m³/h; 30 - 680 m TH; 1800 - 3600 rpm

Typical Application: Loading, transfer, pipeline booster, boiler feed

Courtesy of Bingham – Willamette Ltd.

FIG. 12-6e

Vertical, High Pressure, Double Case, Multi-Stage Pump



Pump Type: Vertical, double case, high pressure, multi-stage barrel type

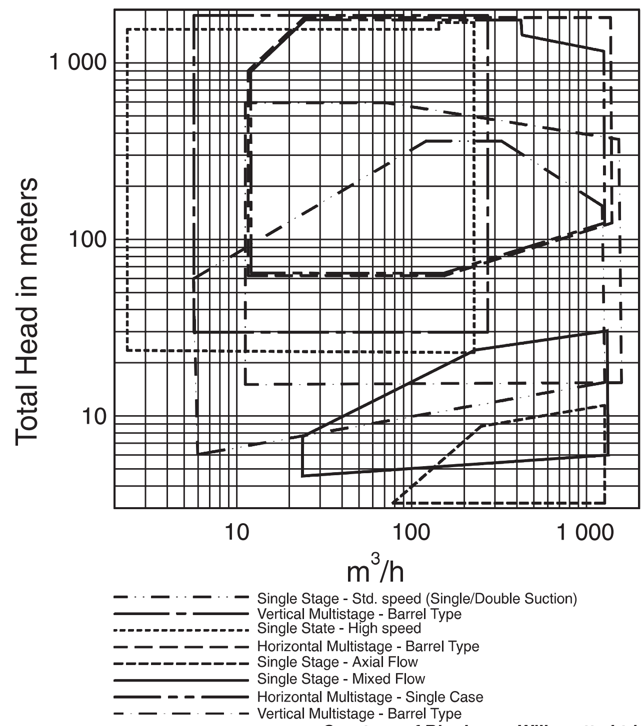
11 - 230 m³/h; 150 - 2440 m TH; 3600 rpm

Typical Application: High pressure injection, ethane product, miscible flood, boiler feed

Courtesy of Bingham – Willamette Ltd.

FIG. 12-7

Pump Selection Guide — Centrifugal Pumps



Courtesy of Bingham—Willamette Ltd.

— The head loss expected to result from each component which creates a frictional pressure drop at design capacity.

3. Use appropriate equations (Eq 12-1 to Eq 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.
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. In order to provide good control, a discharge control valve should be designed to absorb at least 30% of the frictional head loss of the system, at the design flow rate.
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.12-9. The pump curves are shown in Fig. 12-5. The pump nozzles elevations are zero and the velocity head at nozzles is negligible.

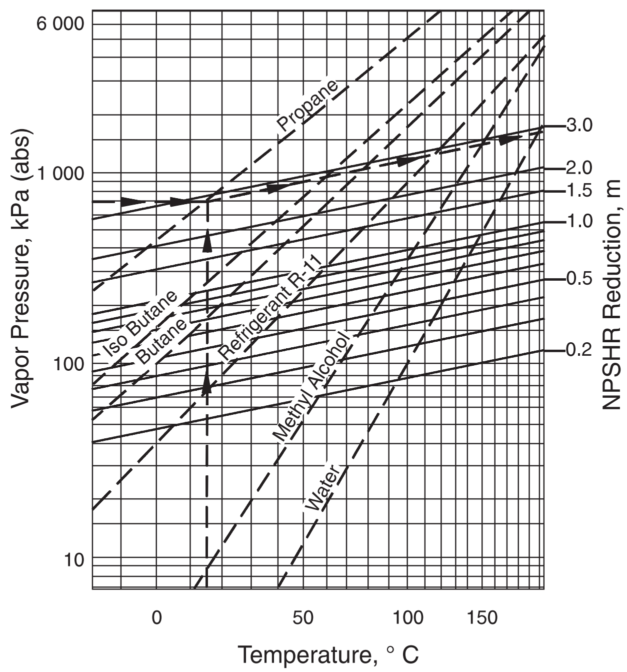
Required differential head is determined as follows:

Absolute Total Pressure at Pump Suction

Reflux drum		1380 kPa (abs)
Elevation	$6m \cdot 0.999 \cdot 0.485 \cdot 9.807$	= +28.5 kPa
Friction	piping	−3.5 kPa
	valves	−1.4 kPa
		1403.6 kPa (abs)
		= 1302.3 kPa (ga)

FIG. 12-8

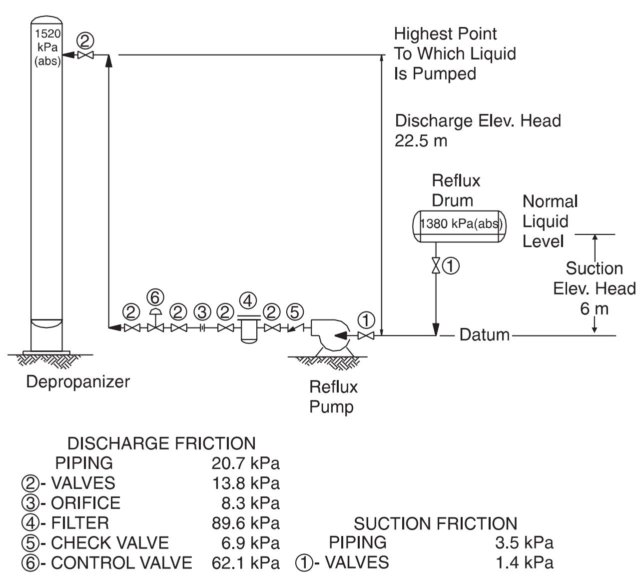
NPSHR Reduction for Centrifugal Pumps Handling Hydrocarbon Liquids and High Temperature Water



This chart has been constructed from test data using the liquids shown. For applicability to other liquids refer to the Hydraulic Institute Standards.[5]

FIG. 12-9

Example 12-1 Depropanizer



Absolute Total Pressure at Pump Discharge

Tower			1520 kPa (abs)
Elevation	22.5 m • 0.999 • 0.485 • 9.807	=	+106.9 kPa
Friction	piping		+20.7 kPa
	valves		+13.8 kPa
	orifice		+8.3 kPa
	filter		+89.6 kPa
	check valve		+6.9 kPa
	control valve		+62.1 kPa
			1828.3 kPa (abs)
			= 1727.0 kPa (ga)

Differential pressure = 1727 – 1302.3 = 424.7 kPa

$$\text{Differential head} = H = \frac{(424.7)}{(0.485)(0.999)(9.807)} = 89.4 \text{ m}$$

10% safety factor	9 m
Required differential head (H)	98.4 m

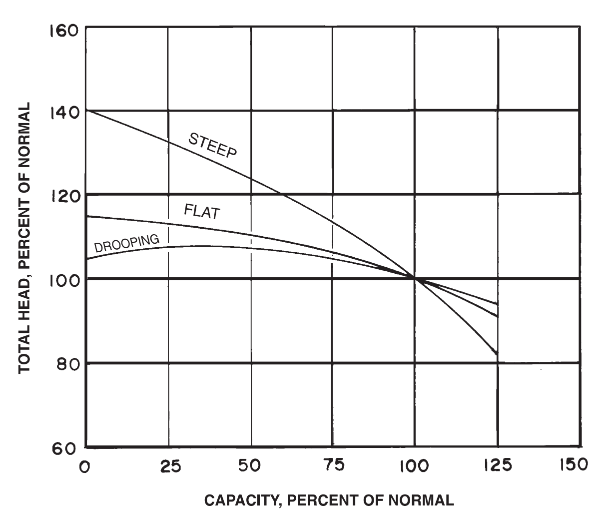
Calculation of NPSHA

Reflux drum pressure			1380 kPa (abs)
Elevation	6 m • 0.999 • 0.485 • 9.807	=	+28.5 kPa
Friction	valves =		–1.4 kPa
	piping =		–3.5 kPa
Fluid vapor pressure			–1380 kPa (abs)
			23.6 kPa
NPSHA	23.6/(0.999 • 0.485 • 9.807)	=	5.0 m

This NPSHA result is adequate when compared to the 3 m of NPSHR in the curve shown in Fig. 12-5.

FIG. 12-10

Example Centrifugal Pump Head Curves



Calculation of Hydraulic Power

$$\text{hyd kW} = \frac{Q \bullet H \bullet RD}{367} \qquad \textbf{(from Fig. 12-2)}$$

$$\text{hyd kW} = \frac{(82)\,(98.4)\,(0.485)}{367} = 10.67 \text{ kW}$$

Calculation of Actual Horsepower

$$\text{bkW} = \frac{\text{hyd kW}}{e} \qquad \textbf{(from Fig. 12-2)}$$

Fig. 12-5 is the performance curve of the selected pump. The efficiency at rated capacity and required head is 62%, with a brake kilowatt calculated as follows:

$$\text{bkW} = \frac{10.67 \text{ kW}}{0.62} = 17.2 \text{ bkW}$$

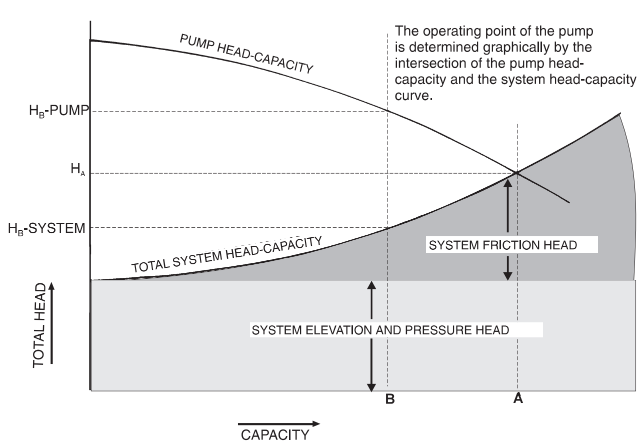
Motor Sizing — The maximum flow is 115 m^3/h with a head of 75 m for this particular pump impeller size, which results in a brake kilowatt requirement of 19.5 bkW at run-out (i.e., end of head curve). Therefore a 25 kW motor is selected for the pump driver to provide "full curve" protection.

CENTRIFUGAL PUMPS

Figs. 12-6a through 12-6e are cross-sectional drawings showing typical configurations for five types of centrifugal pumps. A guide to selecting centrifugal pumps is shown in Fig. 12-7. Horizontal centrifugal pumps are more common; however, vertical pumps are often used because they are more compact and, in cold climates, may need less winterizing than horizontal pumps. The total installed cost of vertical pumps is frequently lower than equivalent horizontal pumps because they require smaller foundations and simpler piping systems.

FIG. 12-11

Example Combined Pump-System Curves



Vertical can pumps are often used for liquids at their bubble-point temperature because the first stage impeller is located below ground level and therefore requires less net positive suction head at the suction flange. The vertical distance from the suction flange down to the inlet of the first stage impeller provides additional NPSHA.

Centrifugal Pump Theory

Centrifugal pumps increase the pressure of the pumped fluid by action of centrifugal force on the fluid. Since the total head produced by a centrifugal pump is independent of the density of the pumped fluid, it is customary to express the pressure increase produced by centrifugal pumps in feet of head of fluid pumped.

Operating characteristics of centrifugal pumps are expressed in a pump curve similar to Fig. 12-5. Depending on impeller design, pump curves may be "drooping," "flat," or "steep." Fig. 12-10 shows these curves graphically. Pumps with drooping curves tend to have the highest efficiency but may be undesirable because it is possible for them to operate at either of two flow rates at the same head. The influence of impeller design on pump curves is discussed in detail in Hydraulic Institute Standards.[5]

Affinity Laws for Centrifugal Pumps — The relationships between rotational speeds, impeller diameter, capacity, head, power, and NPSHR for any particular pump are defined by the affinity laws (See Fig. 12-2 for affinity laws). These equations are to predict new curves for changes in impeller diameter and speed.

The capacity of a centrifugal pump is directly proportional to its speed of rotation and its impeller diameter. The total pump head developed is proportional to the square of its speed and its impeller diameter. The power consumed is proportional to the cube of its speed and its impeller diameter. The NPSHR is proportional to the square of its speed.

These equations apply in any consistent set of units but only apply exactly if there is no change of efficiency when the rotational speed is changed. This is usually a good approximation if the change in rotational speed is small.

A different impeller may be installed or the existing modified. The modified impeller may not be geometrically similar to the original. An approximation may be found if it is assumed that the change in diameter changes the discharge peripheral velocity without affecting the efficiency. Therefore, at equal efficiencies and rotational speed, for small variations in impeller diameter, changes may be calculated using the affinity laws.

These equations do not apply to geometrically similar but different size pumps. In that case dimensional analysis should be applied.

The affinity equations apply to pumps with radial flow impellers, that is, in the centrifugal range of specific speeds, below 4200. For axial or mixed flow pumps, consult the manufacturer. See Fig. 12-2 for specific speed equation.

Viscosity

Most liquids pumped in gas processing plants have viscosities in the same range as water. Thus they are considered "nonviscous" and no viscosity corrections are required. Occasionally fluids with viscosities higher than 5 x 10^{-6} m^2/s are encountered (e.g. triethylene glycol, 40 x 10^{-6} m^2/s at 20°C) and corrections to head, capacity, and power consumption may be required.

Viscosity correction charts and the procedures for using them are included in Hydraulic Institute Standards.[5]

Matching the Pump to the System Requirements

A pump curve depicts the relationship between the head and capacity of a pump. A system curve shows the relationship between the total head difference across the system and the flow rate through it. The total head difference consists of three components: static (gravity) head, pressure head, and head-loss due to friction. Static and pressure heads do not change with flow. However, frictional losses usually increase approximately as the square of the flow rate through the system. If the system curve is plotted with the same units as the pump curve, it can be superimposed as shown in Fig. 12-11.

For pump selection, the shape and slope of the pump curve shall be considered in its position with respect to the system curve. When the curves are approximately perpendicular to each other, the change in the operating point position due to deviations in the curves will be minimum. In addition, the shape and slope shall be considered when several pumps are used in series and/or parallel operation to produce the desired range of flow and/or operating pressure. Refer to Fig. 12-12 and Fig. 12-13 for series and parallel operation.

Throttling Control — If a centrifugal pump and a system were matched as shown in Fig. 12-11, the flow rate through the system will be "A" unless some kind of flow control is provided. Control usually is provided by throttling a valve in the discharge piping of the pump, which creates extra frictional losses so that pump capacity is reduced to that required. In Fig. 12-11, the required flow rate is represented by "B." Required amount of extra frictional losses to achieve a flow rate of "B" is represented on Fig. 12-11 by the difference between "H_B-PUMP" and "H_B-SYSTEM." Frequently the throttling valve is an automatic control valve which holds some plant condition constant (such as liquid level, flow rate, or fluid temperature). This control method consumes energy since it artificially increases the system resistance to flow.

Recirculation Control — Pump capacity can also be controlled by recirculating a portion of the pumped fluid back to the suction. This control method is used more frequently for positive displacement pumps than for centrifugal pumps, since the discharge of most positive displacement pumps should not be throttled. This control method should be used with caution for centrifugal pumps, since a wide-open recirculation may result in a head so low that the pumped fluid will be circulated back to the suction at an extremely high rate, causing high power consumption, increase in fluid temperature, and possibly cavitation, as well as possibly overloading the driver.

Speed Control — Another way of regulating centrifugal pump capacity is to adjust the rotational speed of the pump. This is frequently not easily done because most pumps are driven by fixed-speed motors. However, pumps controlled by adjusting the rotational speed often consume substantially less energy than those controlled in other ways. The changed power consumption can be calculated by Eq. 12-8, which assumes that the frictional head is proportional to the square of the flow rate.

$$bkW_2 = bkW_1\left(\frac{e_1}{e_2}\right)\left[\frac{h_s\,(Q_2/Q_1) + h_{f1}\,(Q_2/Q_1)^3}{h_s + h_{f1}}\right] \qquad \textbf{Eq 12-8}$$

subscript 1 refers to initial flow rate

subscript 2 refers to the changed flow rate

h_s (static) is equivalent to the zero flow system total head

On-Off Control — Pump capacity can be controlled by starting and stopping the pump manually or by an automatic control such as pressure, level or temperature switches.

Temperature Rise Due to Pumping

When a liquid is pumped, its temperature increases because the energy resulting from the inefficiency of the pump appears as heat.

$$t_r = \frac{9.8067 \bullet H\left(\frac{1}{e} - 1\right)}{C_p} \qquad \textbf{Eq 12-9}$$

Usually when the pump is running normally, the temperature rise is negligible. However, if the pump discharge is shut off, all energy is converted to heat and since there is no fluid flow through the pump to carry the heat away, the liquid in the pump will heat rapidly and eventually vaporize. This can produce catastrophic failures, particularly in large multistage pumps.

Pump vendors should be requested to provide data on minimum flow.

Expensive pumps, such as large multistage units, can be protected by installing minimum flow recirculation which will ensure an adequate flow through the pump.

Series and Parallel Operation

Often pumps are installed in series or in parallel with other pumps. In parallel, the capacities at any given head are added; in series, the heads at any given capacity are added. A multi-stage pump is in effect a series of single stage units. Figs. 12-12 and 12-13 show series and parallel pumps curves, a system curve, and the effect of operating one, two or three pumps in a system. In both figures, the operating points for both pumps "A" and "B" are the same only when one pump is operating. For 2 or 3 pumps operating, the points are not the same because of the pump curve shapes. Hence, due consideration should be given to the pump curve shape when selecting pumps for series or parallel operation.

Parallel operation is most effective with identical pumps; however, they do not have to be identical, nor have the same shut-off head or capacity to be paralleled. When pumps are operating in parallel it is imperative that their performance curves rise steadily to shut-off. A drooping curve gives two possible points of operation, and the pump load may oscillate between the two causing surging.

Drivers

Most pumps used in gas processing service are driven by electric motors, usually fixed speed induction motors.

API Standard 610, Section 3.1.4. (Drivers), states:

"Motors shall have power ratings, including the service factor (if any), at least equal to the percentages of power at pump rated conditions given in. . ." the next table. "However, the power at rated conditions shall not exceed the motor name-

FIG. 12-12

Series Pumps Selection

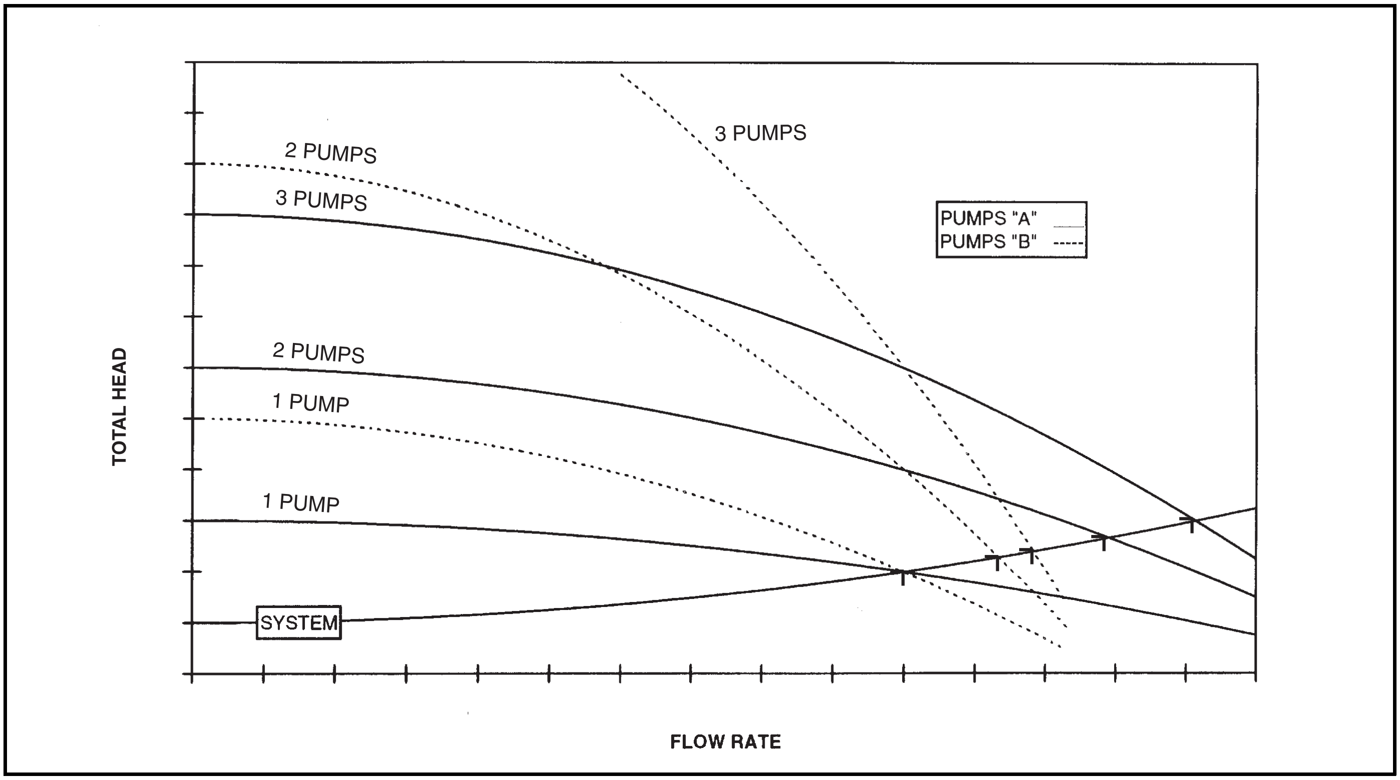


FIG. 12-13

Parallel Pumps Selection

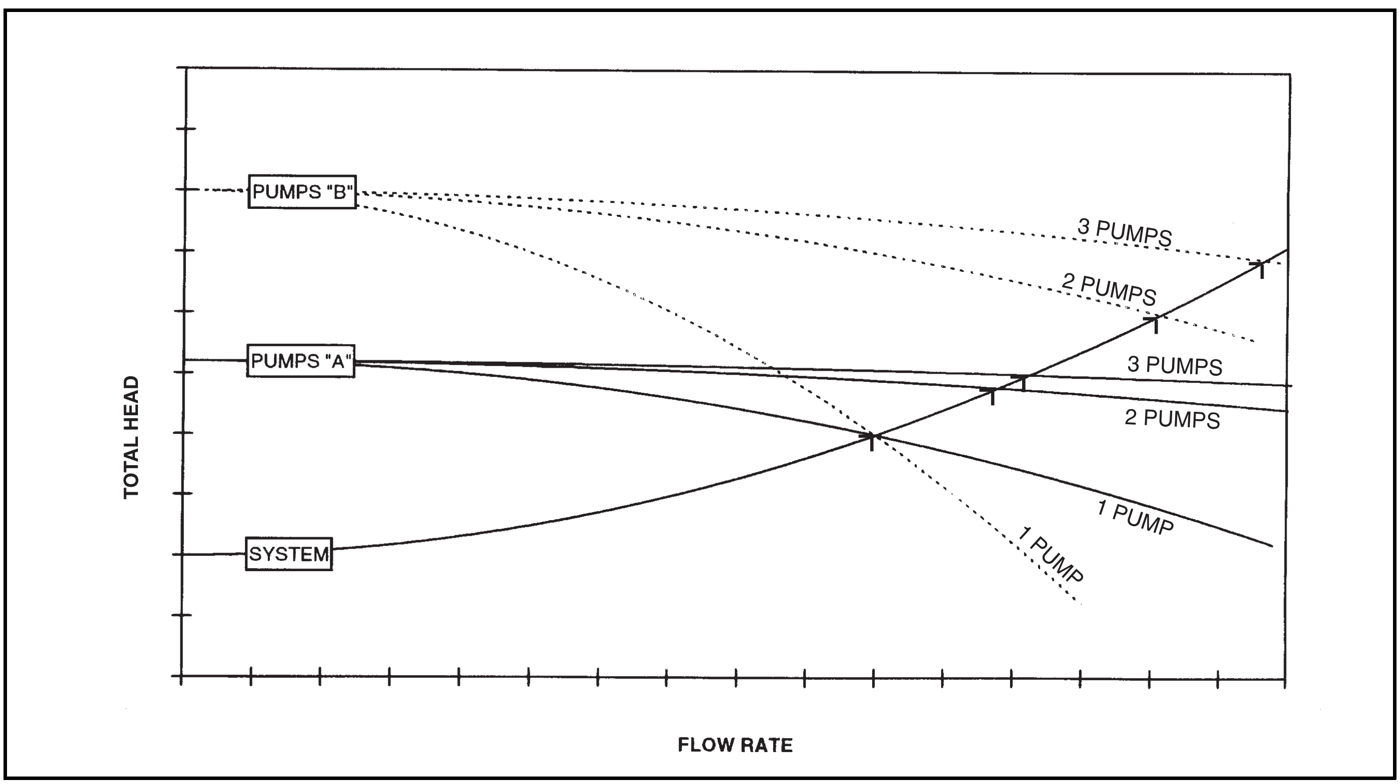


FIG. 12-14

Check List for Centrifugal Pump Troubles and Causes

Trouble:	Possible Causes:
1. Failure to deliver liquid	a. Wrong direction of rotation b. Pump not primed c. Suction line not filled with liquid d. Air or vapor pocket in suction line e. Inlet to suction pipe not sufficiently submerged f. Available NPSH not sufficient g. Pump not up to rated speed h. Total head required greater than head which pump is capable of delivering
2. Pump does not deliver rated capacity	a. Wrong direction of rotation b. Suction line not filled with liquid c. Air or vapor pocket in suction line d. Air leaks in suction line or stuffing boxes e. Inlet to suction pipe not sufficiently submerged. f. Available NPSH not sufficient g. Pump not up to rated speed h. Total head greater than head for which pump designed j. Foot valve too small k. Foot valve clogged with trash m. Viscosity of liquid greater than that for which pump designed n. Mechanical defects: (1) Wearing rings worn (2) Impeller damaged (3) Internal leakage resulting from defective gaskets o. Discharge valve not fully opened
3. Pump does not develop rated discharge pressure	a. Gas or vapor in liquid b. Pump not up to rated speed c. Discharge pressure greater than pressure for which pump designed d. Viscosity of liquid greater than that for which pump designed e. Wrong rotation f. Mechanical defects: (1) Wearing rings worn (2) Impeller damaged (3) Internal leakage resulting from defective gaskets
4. Pump loses liquid after starting	a. Suction line not filled with liquid b. Air leaks in suction line or stuffing boxes c. Gas or vapor in liquid d. Air or vapor pockets in suction line e. Inlet to suction pipe not sufficiently submerged f. Available NPSH not sufficient g. Liquid seal piping to lantern ring plugged h. Lantern ring not properly located in stuffing box

Trouble:	Possible Causes:
5. Pump overloads driver	a. Speed too high b. Total head lower than rated head c. Excessive recirculation d. Either or both the relative density and viscosity of liquid different from that for which pump is rated e. Mechanical defects: (1) Misalignment (2) Shaft bent (3) Rotating element dragging (4) Packing too tight
6. Vibration	a. Starved suction (1) Gas or vapor in liquid (2) Available NPSH not sufficient (3) Inlet to suction line not sufficiently submerged (4) Gas or vapor pockets in suction line b. Misalignment c. Worn or loose bearings d. Rotor out of balance (1) Impeller plugged (2) Impeller damaged e. Shaft bent f. Improper location of control valve in discharge line g. Foundation not rigid
7. Stuffing boxes overheat	a. Packing too tight b. Packing not lubricated c. Wrong grade of packing d. Insufficient cooling water to jackets e. Box improperly packed.
8. Bearings overheat	a. Oil level too low b. Improper or poor grade of oil c. Dirt in bearings d. Dirt in oil e. Moisture in oil f. Oil cooler clogged or scaled g. Failure of oiling system h. Insufficient cooling water circulation i. Insufficient cooling air k. Bearings too tight m. Oil seals too close fit on shaft n. Misalignment
9. Bearings wear rapidly	a. Misalignment b. Shaft bent c. Vibration d. Excessive thrust resulting from mechanical failure inside the pump e. Lack of lubrication f. Bearings improperly installed g. Dirt in bearings h. Moisture in oil j. Excessive cooling of bearings

plate rating. Where it appears that this procedure will lead to unnecessary oversizing of the motor, an alternate proposal shall be submitted for the purchaser's approval."

Motor Nameplate Rating		Percentage of Rated Pump Power
kW	hp	
<22	<30	125
22-55	30-75	115
>55	>75	110

Alternatives to electric motor drivers are:

- internal combustion engines
- gas turbines
- steam turbines
- hydraulic power-recovery turbines

Usually the speed of rotation of these drivers can be varied to provide control.

Variable Speed Drives — Fig. 12-15 lists various types of adjustable speed drives, their characteristics and their application.

Materials of Construction

Pumps manufactured with cast-steel cases and cast-iron internals are most common in the gas processing industry. API Std 610 is a good reference for material selection. The material selections in this document can be over-ridden as required to reflect experience.

Experience is the best guide to selection of materials for pumps. Process pump manufacturers can usually provide suggestions for materials, based on their experience and knowledge of pumps.

Shaft Seals

Mechanical seals are the most common sealing devices for centrifugal pumps in process service. The purpose of the seal is to retain the pumped liquid inside the pump at the point where the drive shaft penetrates the pump body. Mechanical seals consist of a stationary and a rotating face, and the actual sealing takes place across these very smooth, precision faces. Seal faces may require cooling and lubrication. API Std 610 describes seal flush systems used to cool the seal faces and remove foreign material. Seal manufacturers can provide application and design information.

Alignment, Supports, and Couplings

The alignment of the pump and driver should be checked and adjusted in accordance with the manufacturer's recommendations before the pump is started. If the operating temperature is greatly different from the temperature at which the alignment was performed, the alignment should be checked, and adjusted if necessary, at the operating temperature.

Pump and piping supports should be designed and installed so that forces exerted on the pump by the piping will not cause pump misalignment when operating temperature changes or other conditions occur.

The shaft coupling should be selected to match the power transmitted and the type of pump and driver. A spacer type coupling should be used if it is inconvenient to move either the pump or the driver when the seal (or other component) requires maintenance.

Piping

Pump requirements, nozzle size, type of fluid, temperature, pressure and economics determine materials and size of piping.

Suction lines should be designed to keep friction losses to a minimum. This is accomplished by using an adequate line size, long radius elbows, full bore valves, etc. Pockets where air or vapor can accumulate should be avoided. Suction lines should be sloped, where possible, toward the pump when it is below the source, and toward the source when it is below the pump. Vertical downward suction pipes require special care to avoid pulsation and vibrations that can be caused by air or vapor entrainment. Elbows entering double suction pumps should be installed in a position parallel to the impeller. Sufficient liquid height above the suction piping inlet, or a vortex breaker, should be provided to avoid vortex formation which may result in vapors entering the pump.

For discharge piping, sizing is determined by the available head and economic considerations. Velocities range from 1 to 5 m/s. A check valve should be installed between the discharge nozzle and the block valve to prevent backflow.

Auxiliary piping (cooling, seal flushing and lubrication) is a small but extremely important item. API Standard 610, "Centrifugal Pumps for General Refinery Service," or applicable national standard should be followed. Provisions for piping of stuffing box leakage and other drainage away from the pump should be provided.

Pump Protection

The following protection may be considered:

- low suction pressure
- high discharge pressure
- low suction vessel (or tank) level
- high discharge vessel (or tank) level
- low flow
- flow reversal
- high temperature of bearings, case, etc.
- vibrations
- lack of lubrication
- overspeed

Protection may be considered for the pump driver and may be combined with pump protections.

Installation, Operation, Maintenance

Installation, operation, and maintenance manuals should be provided by the pump manufacturer and are usually application specific. See Fig. 12-14 for a checklist of pump troubles and causes.

Driver rotation and alignment should be checked before the pump is operated.

A typical starting sequence for a centrifugal pump is:

- Ensure that all valves in auxiliary sealing, cooling, and flushing system piping are open, and that these systems are functioning properly.
- Close discharge valve.
- Open suction valve.
- Vent gas from the pump and associated piping.
- Energize the driver.
- Open discharge valve slowly so that the flow increases gradually.

- Note that, on larger multistage pumps, it is very important that flow through the pump is established in a matter of seconds. This is frequently accomplished by the previously mentioned minimum flow recirculation.

RECIPROCATING PUMPS

The most common reciprocating pump in gas plants is the single-acting plunger pump which is generally employed in services with moderate capacity and high differential pressure. These pumps fill on the backstroke and exhaust on the forward stroke. They are available with single (simplex) or multi-plungers (duplex, triplex, etc.), operating either horizontally or vertically. Examples of plunger pump service in gas plants are: high pressure chemical or water injection, glycol circulation, and low capacity, high pressure amine circulation, and pipeline product pumps.

Double-acting piston pumps which fill and exhaust on the same stroke have the advantage of operating at low speeds and can pump high viscosity liquids which are difficult to handle with normal centrifugal or higher speed plunger pumps.

Pump Calculations

Power requirement bkW: see equation in Fig. 12-2.

Displacement for single-acting pump

$$D = 60 \bullet 10^{-9} \bullet A \bullet m \bullet L_s \bullet n \qquad \textbf{Eq 12-10}$$

Displacement for double-acting pump

$$D = 60 \bullet 10^{-9} \bullet (2 \bullet A - a)\ m \bullet L_s \bullet n \qquad \textbf{Eq 12-11}$$

Notes:

1. Actual capacity (Q) delivered by pump is calculated by multiplying displacement by the volumetric efficiency.
2. The combination of mechanical and volumetric efficiency for reciprocating pumps is normally 90% or higher for noncompressible fluids.
3. In double-acting pumps with guided piston (rod in both sides), change “a” to “2a” in Eq 12-11.

Example 12-2 — Calculate the power required for a simplex plunger pump delivering 2.3 m^3/h of liquid of any relative density at 20 000 kPa differential pressure and mechanical efficiency of 90%.

FIG. 12-15

Adjustable Speed Drives[3] and Power Transmissions

Type	Characteristics	Applications
Electric Drivers		
Solid State AC drives	• high efficiency • good speed regulation • low maintenance • complex controls • high cost • can be explosion proof • can retrofit	• 50 to 2500+ bhp • larger pumps where good speed regulation over not too wide a range is required • hazardous areas
Solid State DC drives	• similar to AC except speed regulation good over a wider range	• 50 to 500+ bhp • non-hazardous areas
Electromechanical		
Eddy Current Clutch	• efficient, proportional to slip • poor speed regulation • require cooling	• 5 to 500+ bhp • smaller centrifugal pumps where speed is usually near design • non-hazardous areas
Wound-Rotor Motor	• poor speed regulation • reasonable efficiency	• 50 to 500+ bhp • larger pumps non-hazardous areas
Mechanical		
Rubber Belt	• wide range of speed regulation possible	• fractional to 100 bhp • small centrifugal and positive displacement pumps
Metal Chain	• low to medium efficiency	• chemical feed pumps • non-hazardous areas
Hydraulic Power Recovery Turbines	• medium efficiency • continuously variable speed • reversible use as pump	• available hydraulic head

$$bkW = \frac{(2.3)\ (20\ 000)}{(3600)\ (0.90)} = 14.2\ kW$$

Volumetric Efficiency, Compressible Fluids — Unlike water, lighter hydrocarbon liquids (e.g. ethane, propane, butane) are sufficiently compressible to affect the performance of reciprocating pumps.

The theoretical flow capacity is never achieved in practice because of leakage through piston packing, stuffing boxes, or valves and because of changes in fluid density when pumping compressible fluids such as light hydrocarbons.

The ratio of real flow rate to theoretical flow rate (pump displacement) is the volumetric efficiency. The volumetric efficiency depends on the size, seals, valves and internal configuration of each pump, the fluid characteristics and operating conditions.

When pumping compressible liquids, the volumetric efficiency should be stated with reference to the flow rate measured in a specific side of the pump (suction or discharge side).

The relationship of overall suction and discharge volumetric efficiency, displacement, and suction and discharge flow rate of a reciprocating pump is defined in Eq 12-12. When the leakage is not considered, the overall efficiencies may be substituted by the density change efficiencies.

$$D = \frac{Q_s}{VE_{sov}} = \frac{Q_d}{VE_{dov}} \qquad \textbf{Eq 12-12}$$

The following equations are based on the discharge flow rate. Similar equations may be written for the suction side, and conversions may be made by multiplying them by the discharge to suction densities ratio.

The overall discharge volumetric efficiency is a combination of volumetric efficiency due to leakage and discharge volumetric efficiency due to fluid density change.

$$VE_{dov} = VE_l \bullet VE_{d\rho} \qquad \textbf{Eq 12-13}$$

The volumetric efficiency due to leakage is related to slip as follows:

$$VE_l = 1 - s \qquad \textbf{Eq 12-14}$$

The effect of the difference in the leakage flow rate measured at suction pressure vs discharge pressure is neglected here, assuming that all leakages are internal.

The discharge volumetric efficiency due to density change is:

$$VE_{d\rho} = 1 - r\left[1 - \frac{\rho_i}{\rho_o}\right] \qquad \textbf{Eq 12-15}$$

When the change in fluid density is linear with the change in pressure and is smaller than 10%, and the temperature change is negligible, Equation 12-16 may be used to calculate hydraulic power. H_c comes from Eq 12-5. Additionally, approximately 2 to 5% of power may be required for the work done during the piston cycle, in compressing and in decompressing the fluid that is held in the pump chamber without flowing through the pump.

$$hyd\ kW = \frac{Q_d \bullet \rho_o \bullet g \bullet H_c}{3\ 600\ 000} \qquad \textbf{Eq 12-16}$$

When the differential pressure is sufficiently high to cause a density change of more than 10%, or when the pressure is near the fluid's critical pressure, or when temperature change is not negligible, this equation may not be accurate. In such cases the pump manufacturer should be consulted. See Equipment and System Equations last paragraph.

Data on density change with pressure and temperature can be found in Section 23, "Physical Properties."

Example 12-3 — For a 75 mm diameter and a 125 mm stroke triplex plunger pump pumping propane with a suction density 505 kg/m^3 and a discharge density 525 kg/m^3 and given that r = 4.6 and s = 0.03, find the overall discharge volumetric efficiency.

Discharge volumetric efficiency due to density change:

$$VE_{d\rho} = 1 - 4.6\left(1 - \frac{505}{525}\right) = 0.824$$

Volumetric efficiency due to leakage:

$$VE_l = 1 - 0.03 = 0.97$$

Overall discharge volumetric efficiency:

$$VE_{dov} = (0.824) \bullet (0.97) = 0.799$$

Suction System Considerations

The suction piping is a critical part of any reciprocating pump installation. The suction line should be as short as possible and sized to provide not more than three feet per second fluid velocity, with a minimum of bends and fittings. A centrifugal booster pump is often used ahead of a reciprocating pump to provide adequate NPSH which would also allow higher suction line velocities.

FIG. 12-16

Reciprocating Pump Acceleration Head Factors

C = 0.200 for simplex double-acting	k = a factor related to the fluid compressibility		
= 0.200 for duplex single-acting		hot oil	2.5
= 0.115 for duplex double-acting		most hydrocarbons	2.0
= 0.066 for triplex single or double-acting		amine, glycol, water	1.5
= 0.040 for quintuplex single or double-acting		deaerated water	1.4
= 0.028 for septuplex single or double-acting		liquid with small amounts of entrained gas	1.0
= 0.022 for nonuplex single or double-acting			

Note: "C" will vary from the listed values for unusual ratios of connecting rod length to crank radius over 6.

NPSH required for a reciprocating pump is calculated in the same manner as for a centrifugal pump, except that additional allowance must be made for the requirements of the reciprocating action of the pump. The additional requirement is termed acceleration head. This is the head required to accelerate the fluid column on each suction stroke so that this column will, at a minimum, catch up with the receding face of the piston during its filling stroke.

Acceleration Head — Acceleration head is the fluctuation of the suction head above and below the average due to the inertia effect of the fluid mass in the suction line. With the higher speed of present-day pumps or with relatively long suction lines, this pressure fluctuation or acceleration head must be taken into account if the pump is to fill properly without forming vapor which will cause pounding or vibration of the suction line.

With the slider-crank drive of a reciprocating pump, maximum plunger acceleration occurs at the start and end of each stroke. The head required to accelerate the fluid column (h_a) is a function of the length of the suction line and average velocity in this line, the number of strokes per minute (rpm), the type of pump and the relative elasticity of the fluid and the pipe, and may be calculated as follows:

$$h_a = \frac{L \bullet v \bullet n \bullet C}{k \bullet g} \qquad \textbf{Eq 12-17}$$

where C and k are given in Fig. 12-16.

Example 12-4 — Calculate the acceleration head, given a 50 mm diameter x 125 mm stroke triplex pump running at 360 rpm and displacing 16.5 m^3/h of water with a suction pipe made up of 1.2 m of 4" and 6.1 m of 6" standard wall pipe.

Average Velocity in 4" Pipe = 0.56 m/s

Average Velocity in 6" Pipe = 0.25 m/s

Acceleration Head in 4" Pipe

$$h_{a4} = \frac{(1.2)\,(0.56)\,(360)\,(0.066)}{(1.5)\,(9.8067)} = 1.085 \text{ m}$$

Acceleration Head in 6" Pipe

$$h_{a6} = \frac{(6.1)\,(0.25)\,(360)\,(0.066)}{(1.5)\,(9.8067)} = 2.463 \text{ m}$$

Total Acceleration Head

$$h_a = 1.085 + 2.463 = 3.548 \text{ m}$$

Karassik et al[9] recommend that the NPSHA exceed the NPSHR by 20 to 35 kPa for reciprocating pumps.

Pulsation — A pulsation dampener (suction stabilizer) is a device installed in the suction piping as close as possible to the pump to reduce pressure fluctuations at the pump. It consists of a small pressure vessel containing a cushion of gas (sometimes separated from the pumped fluid by a diaphragm). Pulsation dampeners should be considered for the suction side of any reciprocating pump, but they may not be required if the suction piping is oversized and short, or if the pump operates at less than 150 rpm. A properly installed and maintained pulsation dampener should absorb the cyclical flow variations so that the pressure fluctuations are about the same as those that occur when the suction piping is less than 4.5 m long.

Similar pressure fluctuations occur on the discharge side of every reciprocating pump. Pulsation dampeners are also effective in absorbing flow variations on the discharge side of the pump and should be considered if piping vibration caused by pressure fluctuations appears to be a problem. Pulsation dampener manufacturers have computer programs to analyze this phenomenon and should be consulted for reciprocating pump applications over 35 kW. Discharge pulsation dampeners minimize pressure peaks and contribute to longer pump and pump valve life. The need for pulsation dampeners is increased if multiple pump installations are involved.

Ensure that bladder type pulsation dampeners contain the correct amount of gas.

Capacity Control — Manual or automatic capacity control for one pump or several parallel pumps can be achieved by one or a combination of the following methods:

- on-off control
- recirculation
- variable speed driver or transmission
- variable displacement pump

Drivers — Two types of mechanisms are commonly used for driving reciprocating pumps; one in which the power of a motor or engine is transmitted to a shaft and there is a mechanism to convert its rotative movement to alternating linear movement to drive the pumping piston or plunger. In the other type, there is a power fluid, such as steam, compressed air, or gas acting on a piston, diaphragm or bellow linked to the pumping piston or plunger.

Piping — Suction and discharge piping considerations are similar to those for centrifugal pumps. In addition, acceleration head must be included for pipe sizing. For piping materials and thickness selection, pressure pulsations amplitude and fatigue life should be considered.

ROTARY PUMPS

The rotary pump is a positive displacement type that depends on the close clearance between both rotating and stationary surfaces to seal the discharge from the suction. The most common types of rotary pumps use gear or screw rotating elements. These types of positive displacement pumps are commonly used for viscous liquids for which centrifugal or reciprocating pumps are not suitable. Low viscosity liquids with poor lubricating properties (such as water) are not a proper application for gear or screw pumps.

DIAPHRAGM PUMPS

Diaphragm pumps are reciprocating, positive displacement type pumps, utilizing a valving system similar to a plunger pump. These pumps can deliver a small, precisely controlled amount of liquid at a moderate to very high discharge pressure. Diaphragm pumps are commonly used as chemical injection pumps because of their controllable metering capability, the wide range of materials in which they can be fabricated, and their inherent leakproof design.

MULTIPHASE PUMPS

Multiphase pumps can pump immiscible liquids such as oil and water with gas. There are screw types and rotodynamic types. A progressive cavity design is used along the flow path to accommodate gas volume reduction caused by increased pressure. A full range of gas/liquid ratios can be handled. This class of pumps is of interest in applications where conventional pumps and separate compressors with or without separate pipelines are not economically feasible.

LOW TEMPERATURE PUMPS

Two types of centrifugal pumps have been developed for cryogenic applications: the external motor type and the submerged motor type.

External motor type — These pumps are of conventional configuration with a coupled driver and can be single or multi-stage. The pump assembly is usually mounted in a vessel from which it pumps.

Submerged motor type — This type of pump is characterized by being directly coupled to its motor, with the complete unit being submerged in the fluid.

Hydraulic Turbines

Many industrial processes involve liquid streams which flow from higher to lower pressures. Usually the flow is controlled with a throttling valve, hence the hydraulic energy is wasted. Up to 80% of this energy can be recovered by passing the liquid through a hydraulic power recovery turbine (HPRT). To justify the installation of an HPRT, an economic analysis of the power savings versus added equipment and installation costs should be performed.

TYPES OF HPRTs

Two major types of centrifugal hydraulic power recovery turbines are used.

1. *Reaction*—Single or multistage Francis-type rotor with fixed or variable guide vanes.
2. *Impulse*—Pelton Wheel, usually specified for relatively high differential pressures.

HPRTs with Francis-type rotors are similar to centrifugal pumps. In fact, a good centrifugal pump can be expected to operate with high efficiency as an HPRT when the direction of flow is reversed.

The Pelton Wheel or impulse runner type HPRT is used in high head applications. The impulse type turbine has a nozzle which directs the high pressure fluid against bowl-shaped buckets on the impulse wheel. This type of turbines' performance is dependent upon back pressure, while the reaction type is less dependent upon back pressure.

Power Recovered by HPRTs

The theoretical energy which can be extracted from a high pressure liquid stream by dropping it to a lower pressure through an HPRT can be calculated using the hydraulic horsepower. See Fig. 12-2 for bkW equation. Since some of the energy will be lost because of friction, the hydraulic horsepower must be multiplied by the efficiency of the HPRT.

The amount of power recovered by an HPRT is directly proportional to the efficiency rather than inversely proportional as is the case when calculating the power required by a pump. Thus, if a fluid is pumped to a high pressure and then reduced to its original pressure using an HPRT, the proportion of the pumping energy which can be supplied by the HPRT is equal to the efficiencies of the pump and turbine multiplied together. Typically, good centrifugal pumps and good HPRTs have efficiencies of between 70% and 80%. Thus, the HPRT can be expected to provide between 50% and 60% of the energy required for pumping.

Usually the high-pressure liquid contains a substantial amount of dissolved gas. The gas comes out of solution as the liquid pressure drops. This does not cause damage to the HPRT, presumably because the fluid velocity through the HPRT is high enough to maintain a froth-flow regime. The term NPSHR does not apply to HPRTs.

Applications

HPRTs may be used to drive any kind of rotating equipment (e.g. pumps, compressors, fans, electrical generators). The main problems are matching the power required by the driven load to that available from the HPRT and speed control. Both the power producer and the speed can be controlled by:

- throttling the liquid flow, either downstream or upstream from the HPRT
- allowing a portion of the liquid to bypass the HPRT
- adjusting inlet guide vanes installed in the HPRT

Sometimes HPRTs are installed with a "helper" driver. If this is an electric motor, the speed will be controlled by the motor speed.

Typical gas-processing streams for which HPRTs should be considered are:

- Rich sweetening solvents (e.g. amines, etc.)
- Rich absorption oil
- High-pressure crude oil.

The lower limit of the power recovery which can be economically justified with single-stage HPRTs is about 22 kW and with multistage, about 75 kW. HPRTs usually pay out their capital cost in from one to three years.

Frequently, when an HPRT is to be used to drive a pump, both devices are purchased from one manufacturer. This has the advantage of ensuring that the responsibility for the entire installation is assumed by a single supplier.

The available pressure differential across the HPRT is calculated using a technique similar to that used to calculate the differential head of centrifugal pumps.

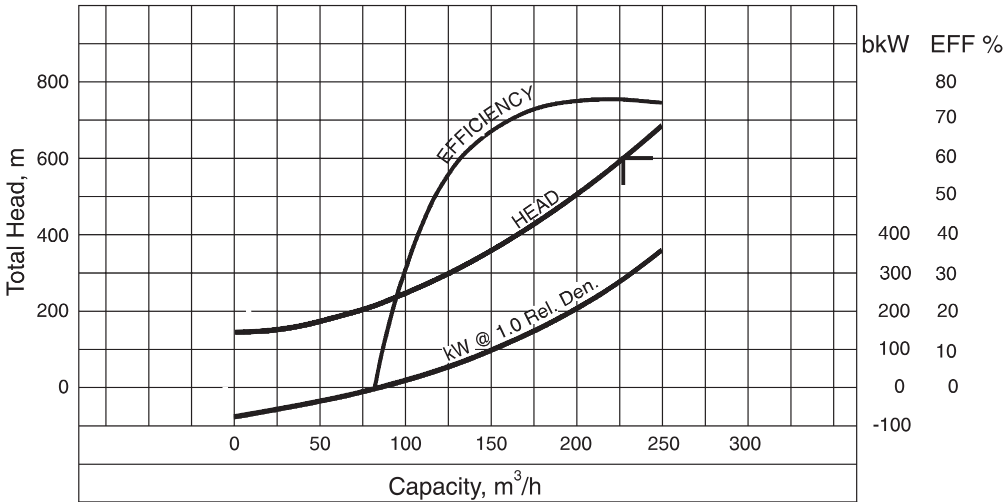
Example 12-5—Specify an HPRT driven pump for a gas sweetening process.

Given:

lean DEA flow	227 m^3/h
lean DEA temperature	43°C

FIG. 12-17

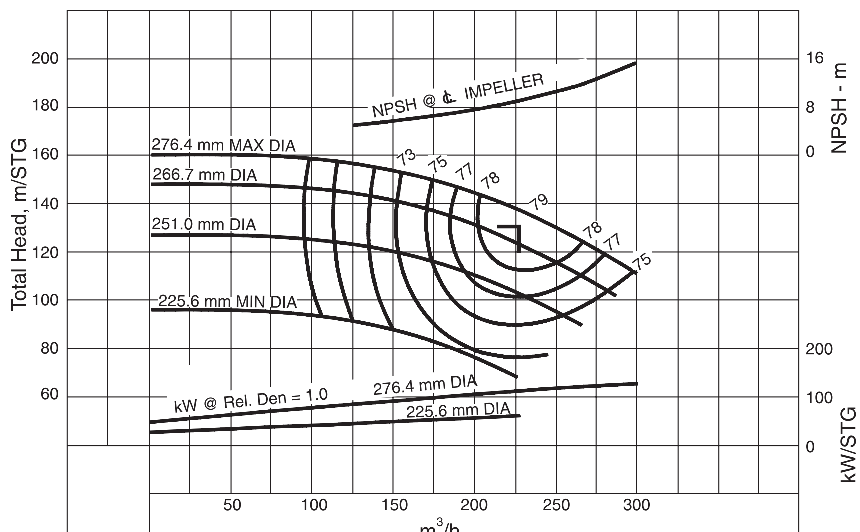
Rich DEA Pressure Letdown



RICH DEA PRESSURE LETDOWN			
IMPELLER MAX DIA 255.5 mm	4 X 6 X 10 HORIZONTAL MULTISTAGE TURBINE		
MIN DIA	DIA IMPELLER 255.5 mm	IMPELLER PATT	3560 RPM
EYE AREA SQ CM	NPSH REQUIRED	REFERENCE	DRAWING NO. HT-3X

FIG. 12-18

Lean Amine Charge Pump



LEAN AMINE CHARGE PUMP			
IMPELLER MAX DIA 276.4 mm	4 X 6 X 10.5 HORIZONTAL MULTISTAGE PUMP		
MIN DIA 225.6 mm	DIA IMPELLER 270 mm	IMPELLER PATT	3560 RPM
EYE AREA 83.9 SQ CM	NPSH REQUIRED	REFERENCE	CURVE NO. MD-38

lean DEA relative density	1.00
lean DEA vapor pressure at 49°C	11.7 kPa (abs)
rich DEA flow	227 m^3/h
rich DEA temperature	71°C
rich DEA relative density	1.01
pump suction total pressure	517 kPa (ga)
pump discharge total pressure	6791 kPa (ga)
HPRT inlet total pressure	6619 kPa (ga)
HPRT outlet total pressure	586 kPa (ga)

Solution:

For this example, the suction and discharge pressures have already been calculated using a technique similar to that suggested for centrifugal pumps.

$$\text{NPSHA for pump} = \frac{(517 + 101.3 - 11.7)}{(9.807)(1.00)(0.999)} = 61.9 \text{ m}$$

$$\text{Required head for pump} = \frac{(6791 - 517)}{(9.807)(1.00)(0.999)} = 640.4 \text{ m}$$

The pump selected is a 5-stage unit. From the pump curve (Fig. 12-18), the expected efficiency of the pump is 78.5%. Hence, the required power will be:

$$\text{bkW for pump} = \frac{(227)(640.4)(1.00)}{(367)(0.785)} = 504.6 \text{ kW}$$

$$\text{Available head for HPRT} = \frac{(6619 - 586)}{(9.807)(1.01)(0.999)} = 609.7 \text{ m}$$

The HPRT selected is a 3-stage unit. From the performance curve (Fig. 12-17), the expected efficiency of the HPRT is 76%. Hence, the available power will be:

$$\text{bkW from HPRT} = \frac{(227)(609.7)(1.01)(0.76)}{367} = 289.5 \text{ kW}$$

Another driver, such as an electric motor, would be required for the pump to make up the difference in bkW between the pump and HPRT. The other driver would have to be capable of providing at least 215 kW. It is good practice to provide an electric motor driver large enough to drive the pump by itself to facilitate startups. The pump, HPRT, and electric motor driver (helper or full size) would usually be direct connected. In some cases, a clutch is used between the pump and HPRT, so the unit is independent of the HPRT.

The pump and HPRT are similar in hydraulic design except that the pump has five stages and the HPRT, three stages. In this case, the HPRT is a centrifugal pump running backwards.

CODES & ORGANIZATIONS

API Std 610 8th Edition—Centrifugal Pumps for General Refinery Service

ANSI B73.1—Horizontal End-Suction Centrifugal Pumps

ANSI B73.2—Vertical Inline Centrifugal Pumps

Hydraulic Institute—Centrifugal, Reciprocating & Rotary Pumps

API Std 674—Positive Displacement Pumps – Reciprocating

API Std 675—Positive Displacement pumps – Controlled Volume

API Std 676—Positive Displacement Pumps – Rotary

API Std 682—Shaft Sealing Systems for Centrifugal and Rotary Pumps.

ANSI/AWWA E101-88—Vertical Turbine Pumps – Line Shaft and Submersible Types

NEMA, EMMAC, UL, CSA—Electric Motor Drivers

UL, ULC, NFPA, FM—Fire Water Pumps

AIChE—American Institute of Chemical Engineers

API—American Petroleum Institute

ANSI—American National Standards Institute

AWWA—American Water Works Association

CSA—Canadian Standards Association

EMMAC—Electrical Manufacturers Association of Canada

FM—Factory Mutual

NEMA—National Electrical Manufacturers Association

NFPA—National Fire Prevention Association

UL—Underwriters Laboratory

ULC—Underwriters Laboratory of Canada

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